



THE GEORGE W. WOODRUFF SCHOOL OF  
MECHANICAL ENGINEERING

Georgia Institute of Technology  
Atlanta, Georgia 30332-0405

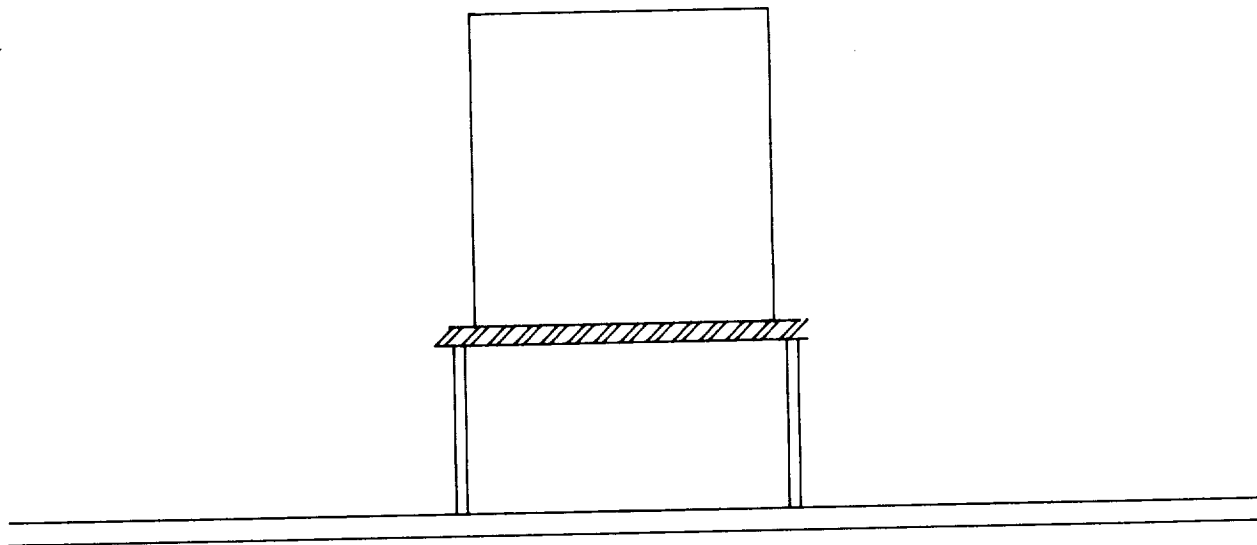
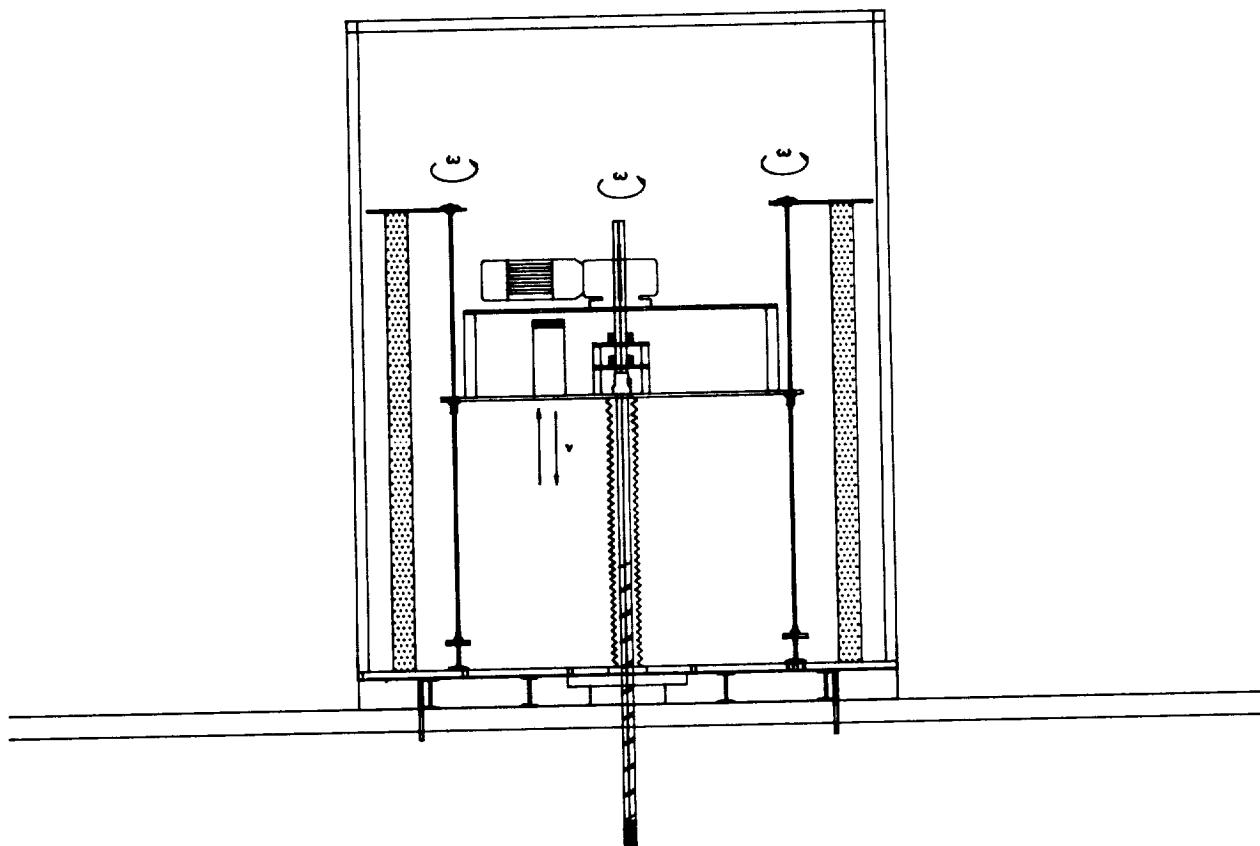
ME 4182  
MECHANICAL DESIGN ENGINEERING

NASA/UNIVERSITY  
ADVANCED DESIGN PROGRAM

LUNAR DRILL AND TEST APPARATUS

NOVEMBER 1988

David W. Norrington  
Didier C. Ardoin  
Stephen G. Alexander  
Philip N. Rowland  
Frank N. Vastakis  
Steven L. Linsey



## TABLE OF CONTENTS

ABSTRACT . . . . .	1
PROBLEM STATEMENT . . . . .	2
BACKGROUND ON PDC TECHNOLOGY . . . . .	3
DESCRIPTION . . . . .	5
ANALYSIS . . . . .	8
PDC CUTTERS . . . . .	8
STUDS . . . . .	8
BIT/AUGER . . . . .	9
AUGER . . . . .	10
BIT/AUGER WALL THICKNESS . . . . .	11
THERMOSYPHON . . . . .	11
THREADED CONNECTIONS . . . . .	12
FORCE, TORQUE, AND POWER REQUIREMENTS . . . . .	13
WEAR CONSIDERATIONS . . . . .	13
DRILL DRIVE MECHANISM . . . . .	14
ROTARY SPEED MEASUREMENT . . . . .	19
THRUST MEASUREMENT . . . . .	20
TORQUE MEASUREMENT . . . . .	22
PENETRATION RATE MEASUREMENT . . . . .	23
TEMPERATURE MEASUREMENT . . . . .	25
CONCLUSIONS AND RECOMMENDATIONS . . . . .	28
ACKNOWLEDGEMENTS . . . . .	31
REFERENCES . . . . .	32

APPENDICES . . . . .34

Appendix 1 Calculations

Appendix 1A Spherical Roller Bearing Calculations

Appendix 1B Screwjack Bearing Calculations

Appendix 1C Screwjack Motor Calculations

Appendix 1D Welded Bellows Calculations

Appendix 1E Single Cutter Data Calculations

Appendix 1F Cutter Stress Calculations

Appendix 1G Bit Face Thickness Calculations

Appendix 1H Buckling of Drill Stem

Appendix 1I Torque and Power Requirements

Appendix 2 Specifications of Drive Mechanism Parts

Appendix 3 Radial Locations of Cutters and Stud Holes

Appendix 4 Tolerances and Fits for Studs and Holes

Appendix 5 Costs

Appendix 6 Progress Reports

Appendix 7 Figures

## ABSTRACT

This report describes the design of an experimental lunar drill and a facility to test the drill under simulated lunar conditions. The drill utilizes a polycrystalline diamond compact drag bit and an auger to mechanically remove cuttings from the hole. The drill will be tested in a vacuum chamber and powered through a vacuum seal by a drive mechanism located above the chamber. The report consists of a general description of the design followed by a detailed description and analysis of each component. Because the design was not completed, the report does not contain a step by step description of a test procedure. The report does include recommendations for the further development of the design.

## PROBLEM STATEMENT

### Background and Objectives

Drilling will be an important part of any future exploration, colonization, or exploitation of the moon. Reliable and effective lunar drilling technologies which may be operated automatically must be developed. Our objective is to contribute to the development of these technologies by designing a drill bit capable of drilling a 51 mm diameter hole into lunar rock and of transporting the cuttings to a chip basket for removal from the hole. We will also design an apparatus to test the bit in a simulated lunar atmosphere using pre-existing vacuum tanks.

### Constraints

The bit must drill rock in a hard vacuum at one sixth of earth gravity without contaminating the walls of the hole or the surrounding lunar environment. The design should be simple, reliable, predictable, and easily maintained since the bit will ultimately be integrated into an automated system. A vacuum must be maintained around the specimen during testing, and the test apparatus must measure rpm, torque, penetration rate, bit temperature, and thrust on bit.

## Background on PDC Technology

Polycrystalline diamond compact (PDC) cutters consist of a polycrystalline man-made diamond substance which is bonded onto a tungsten carbide substrate. These cutters were developed by General Electric and are marketed under the name Stratapax. Typically, these cutters are bonded to a tungsten carbide stud which is then pressed into the drill bit.

PDC cutters drill rock by a process of crushing and fracture initiation and propagation, and single cutter tests have shown that penetration of rock occurs when the normal stresses between cutter and rock exceed the compressive strength of the rock (1). The mean penetrating stress is given by

$$= F/A_w ,$$

where  $F$  is the vertical penetrating force, and  $A_w$  is the area of the wearflat that actually contacts the rock (Figure 1.1). Experiments with multiple cutters have shown that interaction between cutters results in penetration at forces that are significantly lower than those predicted by the above equation (2).

Under certain drilling conditions, high temperatures may develop at the wearflat. If high enough, these temperatures can lead to increased wear of the tungsten carbide backing, and produce unfavorable thermal stress in the polycrystalline diamond layer. It has been shown that these effects produce thermally accelerated wear at wearflat temperatures above 350 C (1). This suggests that bit life can be improved if sufficient heat can be

removed from the cutters during the drilling process.

The literature suggests that bit life can be maximized by designing the bit so that each cutter experiences equal cutter wear (2). Design by this criterion is very difficult because the wear depends on the actual depth of cut, speed, and interaction with other cutters. However, this criterion can be approximated by increasing cutter redundancy as radial distance from the bit center increases (2).



## DESCRIPTION

The lunar drill and test apparatus consists of a downhole portion designed to drill lunar rock , a lunar simulant, a drill drive mechanism, and transducers and instrumentation to measure pertinent drilling data.

### Downhole System

The downhole portion of this design features a single piece combination bit/auger made of high strength steel which is approximately 1 m long and is 50 mm in diameter. The bit portion utilizes polycrystalline diamond compact (PDC) drag cutters mounted on tungsten carbide studs which are pressed into the bit body. The bit body has six channels cut in its outer diameter to provide room for cuttings to move from the bit face to the foot of the auger portion. The auger consists of three flights with 23 turns each at a rake of 15 degrees. The bottom half of the interior of the bit/auger is sealed off and partially filled with mercury to create a gravity fed thermosyphon. The upper half of the interior is used to simulate a removable chip basket. The purpose of this is to test the feasibility of collecting cuttings in a downhole location for removal via a wireline. Cuttings will enter the chip basket through small holes cut in the walls of the bit/auger near the top of the auger flights.

### Lunar Simulant

We have chosen two types of rock for use as lunar simulants. These rocks, Dense Basalt and Hornblende Schist, are identified in the literature on the subject as suitable lunar simulants (3). The basalt has a uniaxial compressive strength of 16.4 kpsi and the schist has a compressive strength of 20.2 kpsi. The rocks will be cut into 2m x 2m x 2m blocks that will weigh 4,673 lb (Basalt) and 5,431 lb (Schist).

### Drill Drive Mechanism

The drive system for this application consists of several major components. The first being the test platform which is a support for all the other equipment. The drill stem passes through this plate as well as through a drive platform located above the test platform. The stem is encompassed in a stretch-type vacuum seal. This seal is anchored to both the test platform and the drive platform. It stretches as the drive platform is raised or lowered. This movement is accomplished by attaching the drive plate to the ball nuts of two ball bearing screw jacks. The screw shafts are rotated by a 1.5 horsepower reversible motor. This drives the ball nuts, drive platform, and drill stem up or down and provides the drilling thrust. On the drive platform, a fluid rotary vacuum seal is used to enclose the space between the stretch seal and the drill stem. Above the drive platform is a motor platform. This supports the drive motor and reducer for the drill stem. The entire drive apparatus

shall be enclosed in a steel housing to protect it from excessive abuse.

### **Transducers and Instrumentation**

The transducers and instrumentation are described in detail in the analysis section.

## ANALYSIS

### PDC Cutters

The PDC cutters used on the bit are shown in Figure 1.2. They are diffusion bonded to tungsten carbide studs which are pressed into the bit body. The eight cutters are mounted with 20 degrees back rake and -2 degrees side rake (Figure 1.1). The back rake assures that the cutters remain in compression, and the side rake will impart an outward component to the velocity of the cuttings, and thereby facilitate their removal from the hole (4).

The radial locations of the cutters are shown in Figure 1.3 and Appendix 3. The redundancy in the outer locations approximates the equal cutter wear criterion and should increase the life of the bit (2). The bit profile in Figure 1.3 shows a cross section of the path that would be traced by the cutters if the bit were rotated once without advancing. The bit profile is flat to maximize the interaction between cutters while holding the number of cutters to eight. The interaction between cutters should assure that the actual penetrating forces required will be less than those predicted by the single cutter model (Appendix 1E ).

### Studs

The tungsten carbide studs are shown Figure 1.2. The hole in the center will allow the working fluid of the thermosyphon to come into close proximity to the cutter wearflats. The studs are made of tungsten carbide - 3% cobalt with a compressive strength

of 615 kpsi and a transverse rupture strength of 170 kpsi (5,6). The stress calculations are shown in Appendix 1F. Because tungsten carbide is a relatively brittle material with a high compressive strength, we assumed that failure would occur at the point where the stud meets the bit and the tensile bending stresses are maximum. Assuming a maximum thrust of 3000 lb<sub>f</sub> and equal thrust per cutter gives a penetrating force of 375 lb<sub>f</sub> per cutter. The literature on the subject indicates that it is reasonable to assume that the drag forces will be equal to the penetrating forces (2). We used a value of 3 as a conservative estimate of the stress concentration factor and an outside diameter of 8.2 mm, and we found that an inside diameter of 4 mm results in a safety factor of 1.78.

#### Bit/Auger

The bit/auger is made from AISI 4140 steel normalized at 1600 F, quenched in oil from 1500 F, and tempered at 1000 F. The bit has six cutting channels extending from the bit face to the foot of the three flight auger. Each auger flight is fed by two channels. The face of the bit is flat and is drilled to accommodate the cutter studs (Figure 1.3).

The radial locations of the stud holes in the bit body are shown in Figure 1.3 and Appendix 3. The holes for the outermost cutter are drilled at an angle of 33.5 degrees from vertical in the plane containing the central axis of the bit and the central

axis of the hole. This allows the outermost cutters to extend 0.5 mm beyond the outside diameter of the bit/auger (Appendix #). This should reduce friction and wear of the bit/auger. Tolerances and fits for the studs and holes are given in Appendix 4. These provide an ANSI FN4 force fit (6).

The dimension labeled X in Figure 1.4 was chosen to provide backing for the stud and to eliminate excessive deflection. We ignored the support provided by the force fit and assumed that the entire load is carried by the material above the stud, and that the mechanics can be modeled as a cantilevered beam under a uniform load (Appendix 1G). Using these assumptions, we found that  $X = 8 \text{ mm}$  results in a maximum deflection of .00001 mm, which is quite acceptable. This dimension is also sufficient to avoid failure by shearing(Appendix 1G).

### Auger

The auger flight geometry is shown in Figures 1.3 and 1.5. The maximum volume of cuttings that can occupy the auger at any one time is  $5.95 \times 10^5 \text{ mm}^3$  (Appendix #). Cuttings will travel up the auger and fall into an internal chip basket through perforations in the bit/auger walls (Figure 1.5). If the bottom of the chip basket is located 450 mm below the perforations, the chip basket volume will be  $2.04 \times 10^5 \text{ mm}^3$ , which is the volume of cuttings generated by drilling approximately 100 mm of rock. This should be sufficient to demonstrate whether or not cuttings

can be made to enter the chip basket.

### **Bit/Auger Wall Thickness**

We chose to make the dimensions of the drill pipe the same as those of the bit/auger walls, and we determined these dimensions through buckling considerations. For simplicity, we ignored the added stiffness due to the auger flights, and treated the auger as if it behaved like a pipe. In this way, the analysis applies to both the auger and the pipe. The calculations are shown in Appendix 1H. With an unsupported length of 4 ft, a maximum load of 3000 lb<sub>f</sub>, a factor of safety of 3, and an outside diameter of 36 mm, the required inside diameter is 32.2 mm. However, this does not account for the perforations in the auger walls that lead to the chip basket. To make up for this, we chose an inside diameter of 24 mm. This resulted in a safety factor of 6.8, ignoring the perforations. We feel that this large safety factor and the fact that the auger flights will add stiffness to the system should assure that the drill string will not buckle at the perforations.

### **Thermosyphon**

Wearflat temperatures greater than 350 C have been shown to cause thermally accelerated wear of PDC cutters and studs (1). Therefore some method must be found to remove heat from the cutting surfaces. We propose that a thermosyphon be incorporated into the bit/auger to accomplish this task (Figure 1.5). To

bring the working fluid into close proximity to the cutting surfaces we have designed the tungsten carbide studs with 4 mm holes along the center lines which extend 16 mm into the studs (Figure 1.2). These holes will match up with similar holes drilled into the bit/auger interior creating a passage for the working fluid. We think that mercury might work as a working fluid, but much analysis is needed to determine the feasibility of this proposal. Potential problems include the possibility of forming vapor pockets inside the studs, compatibility of the mercury with the bit/auger and stud materials, and the transfer of heat from the condenser end of the thermosyphon to the environment (7).

#### **Threaded connections**

We propose that threaded drill pipe connections be used to connect the bit auger, the drill pipe, and the drive shaft. We suggest American Petroleum Institute (API) Rotary Shoulder Connection dimensions. We recommend a V. 0.055 thread form with a 12.5 % taper and a 3.7 mm thread height. This thread will have 6 threads per inch and a pitch of 0.167. This connection must be cut into our existing drill pipe which has an outer diameter of 36 mm and an inner diameter of 24 mm. This connection will give us a thread engagement length of approximately 30 mm which we believe is appropriate.

We have not done a stress analysis or a detailed design on this specification and suggest that this be done before the



project proceeds any further. It is possible that the OD of the bit auger and the drill pipe may have to be increased in the vicinity of the joint to maintain the load bearing capabilities of the drill string.

### **Force, Torque, and Power Requirements**

Appendix 1I shows the torque and power requirement for the test apparatus based on a maximum thrust of 3000 lb<sub>f</sub> and a maximum rotary speed of 200 rpm. In the calculations, we have assumed an equal distribution of the load and a coefficient of drag equal to 1.0 (2). The calculations show that the test apparatus must be capable of transmitting at least 1825 lb<sub>f</sub>-in of torque to overcome the cutter drag forces. However, extra torque will be required to overcome friction between the auger and the walls of the hole. We were unable to predict the magnitude of this extra torque, so we doubled our torque requirements in the hope that the actual torque will be less than this design torque. This gives a torque requirement of 3680 lb<sub>f</sub>-in. At a maximum speed of 200 rpm, the test apparatus must deliver 11.58 horsepower.

### **Wear Considerations**

Frictional wear of the outside diameter of the bit/auger will be a major problem with this design. Thin coatings of titanium nitride (TiN) have been shown to enhance the friction and wear characteristics of cutting tools. In some applications,

wear rates can be decreased by a factor of 10 (8). To reduce the wear rate and frictional forces, we propose to coat the bit/auger with a .003 mm layer of TiN using a physical vapor deposition (PVD) process known as sputter ion plating. PVD methods are preferable to chemical vapor deposition (CVD) methods because they are carried out at temperatures well below the normalization temperature of most steels. This eliminates the need to repeat the heat treatment and eliminates thermal distortions which can occur with the high temperature CVD methods (9).

### Drill Drive Mechanism

The overall purpose of this portion of the apparatus is to provide both rotary and linear motion to the drill stem. In approaching this task, several options were investigated and numerous difficulties were encountered.

After extensive research and discussion the following design was agreed upon (Figure 3.1). It was decided that the drive will be mounted outside the vacuum tank with the drill stem entering through the 8 inch flange at the top of the tank (Figure 3.2). A 14' by 10' by 2" steel plate (Figure 3.3) will be mounted on the flange via 14 1" - 8 threads per inch bolts on a 25" bolt circle and an 8" inner diameter, 32" outer diameter vacuum gasket. This will serve as a test platform to support the drive system. There will be two arch supports between the plate and the tank (Figure 3.4). These will help prevent any undue deformation of the plate.

At the flange, a problem was encountered when we attempted to find a method of maintaining a hard vacuum on the inside of the tank and atmospheric pressure outside. In addition, the pipe entering the flange must be allowed to both rotate and move linearly in and out of the tank. Various vacuum seal manufacturers were contacted in the search for a solution to this problem. Many vacuum seals were available that would allow pure rotary motion and even one that would allow pure linear motion but no seals were available for a combination of these actions. After some consideration, it was decided that an outer pipe should be used to encompass the inner 1 11/16" diameter drill stem pipe. In this approach, a rotary vacuum seal could be anchored to the non-rotating outer pipe. The vacuum seal chosen for this application is a fluid-type with a stainless steel housing (Figure 3.5). It has a maximum speed rating of 2870 RPM, a torque capacity of 12,445 in-lb and a thrust load capacity of 5200 lb. This fluid seal will be mounted directly above the drive platform (Figure 3.1) using 6 - 10 - 32 UNF on a 6.0 inch bolt circle and an o-ring. As this seal is capable of supporting a thrust load, it can also act as a thrust bearing. This type of fluid seal requires a small pump for the fluid and has tube fittings for use with tubing in line with the 4 face mounting holes. Although this pump has not been located, they are commercially available from various suppliers as well as the vacuum seal manufacturer. The linear motion vacuum seal for the outer pipe to the flange is somewhat more complex. It basically

consists of a set of stacked washers welded so that the washers can stretch or compress. This apparatus is known as a welded bellows. It has a 8:3 maximum extended length to maximum compressed length ratio. Therefore, in order to achieve a maximum travel of 8 feet (approx. 2 meters), a welded bellows of 12.8 feet maximum extended length and 4.8 feet maximum compressed length is required. The bellows will have an inner diameter of 2 inches and an outer diameter of 3.5 inches. The lower end of the bellows will mount to the test platform via a 32" flange, 25" bolt circle, and 14 - 1" - 8 UNF bolts and an o-ring to seal that connection. This flange will match directly with the flange on the vacuum tank and will use the same bolts. The upper flange of the bellows will mount directly to the drive platform (Figure 3.6), making the actual outer pipe unnecessary. This mounting will use 6 - 10- 32 UNF bolts on a 6.0 inch bolt circle and another o-ring seal. This mates to the drill stem, and allows both linear and rotary motion in the stem.

The previously mentioned drive platform is made of 4' by 8' by 1" thick hot rolled steel and it is mounted to two ball-bearing screw jacks. To facilitate this mounting, the drive platform has two 2.75 inch holes spaced 89.25 inches apart center to center (Figure 3.7). In addition to these holes, there is a 2 inch bore in the center of the plate to allow the drill stem to pass through. The drive platform will be mounted to the screw jack nuts using 4 - .25" bolts on a 3.25 inch bolt circle. In this way, as the screw jack nuts are raised and lowered, the

platform and drill stem are raised and lowered.

The installation of the ball bearing screw jacks is necessary in order to create the required thrust. For this application there is a need for 3000 lb of downward thrust. The ball-bearing screw jacks chosen have a 2.5 inch nominal diameter screw with a 0.25 inch lead. These screw jacks are capable of a thrust load in excess of 5000 lb. The screws are placed in tension to reduce stress on the screw and minimize vibration. By doing this, the screws can be utilized at higher speeds than would be normally possible. The screw shaft should be machined down to a 2.25 in outside diameter rather than increased to avoid any unnecessary stress concentration. To maintain the tension in the screws, 4 - 2.25" diameter spherical roller bearings, one mounted at each end of each screw, will be used. These bearings will use flange block mountings and will be attached to the test platform and to the jack anchor supports (Figure 3.8). The bearings selected have the required 0.75" inner diameter, and are capable of 6000 lbs of tension assuming a radial load of 4700 lbs. This should yield a life of 1000 hours assuming the most stressful conditions. The bottom set of bearings shall be mounted directly to the test platform, over the 2.75 inch bores for the screws, using flange blocks. The upper set shall mount, using flange blocks, on top of jack anchor plates (Figure 3.1). These plates are to be attached to the enclosure wall and each will have 4 8 by 8 I-beam supports welded between the anchor plates and to the test platform. This will ensure increased

rigidity in the structure. A 1.5 horsepower variable-speed, reversible gear motor is to be mounted on the test platform in between the two screw jacks. This motor (Figures 3.1 and 3.10) will drive the two screws using sprockets and 2 high torque drive (HTD) belts. In this way, the level of the drive platform and drill stem may be adjusted, in addition to applying thrust to the drill stem and bit.

Two other thrust bearings are to be used, in similar conditions, to support the weight of the drill stem. These bearings will be mounted in special brackets just above the drive platform (Figures 3.1 and 3.9). These bearings have a  $1 \frac{11}{16}$  inch inner diameter, and will be capable of withstanding a thrust load of 3000 lbs and a radial load of 1300 lbs. These figures produce a design life of 1000 hours. (Mathematical analysis of all bearings may be found in the appendix). The brackets to anchor these bearings will be manufactured so that flange blocks may be attached without interfering with the drill stem motion.

Another platform, the motor platform (Figure 3.11), shall be located above the drive platform (Figure 3.1) to allow placement of the motor and reducer, known henceforth as the gearmotor. This gearmotor consists of a 15 horsepower NEMA "C" face motor and a 5:1 single worm reducer. This reducer is mounted on a 500 RHC gearframe. The reducer is a ring-base type with a hollow shaft output. (See appendix). The drill stem shall become a double keyway shaft above the motor platform. These "keys" should directly oppose each other along the diameter of the

shaft.

It is recommended that the entire apparatus be enclosed in 10 gauge hot rolled steel. This structure should be 17 feet high to allow for ample movement of the motor and drive platforms and the drill stem. The roof should measure 12 feet in length by 8 feet in width. This enclosure will not only increase the overall rigidity of the structure but it will also keep the equipment and area safe from dust, debris, and the weather. Also the structure should have an 3.5 foot by 8 foot door on one of the smaller walls of the enclosure to facilitate easier access for maintenance during testing and an air circulator should be installed to remove excess heat from the area during operation.

Finally, it is suggested that a remote control panel (see appendix) be located in an existing building convenient to the vacuum tank site. This control panel would allow the operator to adjust speed, thrust and drill height as well as display approximate instantaneous values. This building will also be the location for the instrumentation controls.

#### **Rotary Speed Measurement**

The revolutions per minute of the drill shaft will be measured by a microprocessor tachometer/process time indicator manufactured by Kernco Instruments, Inc. (Model number 1736, Catalog number 21C15). The input sensor used will also be from Kernco Instruments, Inc. (Catalog number 995-SXC). The 21C15 displays units of rate : RPM, MPH, IPS. The 21C15 also displays units of

time per event for process time functions. A built-in memory provides recall of minimum, maximum, and average measurements selectable through a front-panel switch. The RPM range for the 995-SXC photoelectric sensor is 1-1,000. The photoelectric sensor will be mounted on a 1.25 foot stand on the drive platform and under the motor platform. The sensor will be two feet from the front of the drill stem and directly under the motor. A reflective mark will be placed on the drill stem 1.5 feet above the drive platform. The reflective material is provided by Kernco when the photoelectric sensor is purchased. This reflective mark will be the indicator for the photoelectric sensor. A cable will connect the sensor to the microprocessor which will display the measured RPM. The microprocessor will be located in a small booth housing all instrumentations outside of the test facility.

#### Specifications

Display : 5 digits (0.5" LED)  
Accuracy : 0.01%  
Power : 120 VAC 50/60 Hz  
Size : 5.6" x 2.8" 6.2" depth  
Weight : 2 lbs. 7oz.

#### Thrust Measurement

The force being produced by the drill stem on the lunar simulant will be measured by hollow load cells manufactured by



Eaton Corp. A load cell will be placed against each set of bearings on the drive platform above the rotary seal pump. The total force exerted will be the sum of the forces produced on each load cell. The inner diameter of each load cell was specified to be 1 3/4".

Specifications for Model 3336 Load Cell

Nominal Load Limit Capacity : 5000 lbs. or 20000 newtons.

Static Overload Capacity : 150%

Sheer Load Limit : 550 lbs.

Bending Load Limit : 12000 in-lbs.

Torque Load Limit : 1850 in-lbs.

Load Limit Deflection : 0.005 in.

Output at Rated Capacity (mV per V, nominal) :  $\pm 2$

Nonlinearity (of rated output) :  $\pm 0.25\%$

Hysteresis (of rated output) :  $\pm 0.15\%$

Repeatability (of rated output) :  $\pm 0.15\%$

Zero Balance (of rated output) :  $\pm 1.0\%$

Bridge Resistance (ohms nominal) : 350

Temp. Range, compensated (F) : + 70 to + 170

Temp. Range, compensated (C) : + 21 to + 77

Temp. Range, usable (F) : - 65 to + 200

Temp. Range, usable (C) : - 18 to + 93

Temp. Effect on Output (reading per F) :  $\pm 0.002\%$

Temp. Effect on Output (reading per C) :  $\pm 0.0036\%$

Temp. Effect on Zero (reading per F) :  $\pm 0.002\%$

Temp. Effect on Zero (reading per C) :  $\pm 0.0036\%$   
Excitation Voltage, Maximum (volts DC or AC rms) : 20  
Insulation Resistance (megohms @ 50 VDC) : >5000  
Number of Bridges : 1

The display instrumentation for the force will be a Tedeo, Inc. Model AD4321. The two load cells will each have cables connecting them to the digital display in the instrumentation booth.

Specifications for the Tedeo, Inc. Model AD4321

- Resolution to 1 part in 1000
- Drives up to 4 x 350 ohm load cells
- 115/230 VAC or 12 VDC power
- Serial RS-232 output
- Rated output : 2mV/V
- Safe overload : 150% rated output
- Excitation : 10-20 VAC or DC

Torque Measurement

The torque transmitted by the gearmotor to the drill stem will be measured by a torque meter manufactured by the Technical Data Division of General Thermodynamics (Model E). The "E" type torque meter is designed to couple between a drive and a driven

member, operate in any position and sense torque in either direction of rotation. The system consists of two parts, the torque sensor and the meter readout. The sensor will be directly connected to the motor and the output shaft. The model number E-80B-IKPF was chosen based on the upper torque range of 1000 lb-ft.

### Specifications

Accuracy : Torque readings are accurate within 0.2% over entire range.

Dual Range : The lower 1/4 and 1/10 of the range may be expanded to read full scale.

Transducer Calibration : Within 1/2% of full scale.

Material : Aluminum body, steel shaft, ball bearings with packed grease.

Loading : Inline, transmitted torque.

Maximum Safe Load : Twice full scale rating.

### Penetration Rate Measurement

The penetration rate of the bit will be obtained by indirectly measuring the distance the bit has moved and sending this information to a count-time-rate display controller.

The distance and rate will be measured in the following way :

1. A 2.25 in diameter, 1/8" thick disk will be mounted from the motor-side jack anchor plate in direct

rolling contact with the solid shaft tip of the screw jack. This disk will have 16 0.25 in slots cut at regular intervals on the outer diameter of the disk.

2. An opto isolator will be mounted on the anchor jack so that the disk will extend  $3/8$ " inside the LED of the device.
3. Each time a slot is passed through the LED, a pulse is sent to the count-time-rate controller at which time the display is updated.
4. For each complete revolution of the screwjack, the bit will travel  $1/4$ ". For each pulse sent to the controller, the bit will have traveled  $(1/4") / 16 = 1/64" = 0.015625"$ .

The opto isolator will consist of an ECG 3101 LED-Photodiode and a 1" diameter,  $1/8$ " thick disk with specified slots. The LED-Photodiode will transmit a pulse to the controller each time a slot passes the photodiode. The disk will be made of plexiglass painted jet-black to prevent light transmission through the disk. The count-time-rate controller is produced by Kessler-Products, Co. (Model Keptrol)

The unit will display count, time, or rate. The time base can be set for sec., min., or hours. A conversion chart will accompany the controller with the following conversions :

<u>RATE (pulse/time)</u>		<u>RATE (units/time)</u>
r	=	r (in/sec)
r	=	r x 60 (in/min)
r	=	r x 3600 (in/hr)
r	=	r x 2.54 (cm/sec)
r	=	r x 152.4 (cm/min)
r	=	r x 9144 (cm/hr)

### Specifications For Controller

Accuracy : 100% (100 kHz count speed, .015% accurate crystal based timer)

Display : 0.0001 - 99.9999 scaling factor 8 digit alphanumeric display, 0.55 in LED

Operating Power Source: 110 or 220 VAC. or 12 to 27 VDC

Preset outputs : Dual, two 10-Amp relays

Preset functions : Alternate action, latching, or momentary output from 0.1 sec to 9.9 sec

Panel Housing : 7.37 in by 2.5 in, 6.00 in deep

### Temperature Measurement

The task of measuring the bit temperature created many problems due to the design of the bit and drive system. Below is a list of possible ways to measure bit temperature and why they

were not applicable in this situation.

Thermocouples--While thermocouples imbedded in the bit provide a very inexpensive, simplistic way of measuring the bit temperature, they are unable to be used due to the fact that they must have wires running from them to the display panel. There is no possible way to accommodate this in the bit design due to the partial solid shaft and heat pipe restrictions. Another problem presented by thermocouple wiring is that the wires would constantly be twisted by the rotating bit and eventually break.

Optical Temperature Transducers--The optical temperature transducer would be ideal for this application if a line of sight were obtainable to the bit tip. This, however, is impossible due to previously mentioned restrictions.

Simulant Imbedded Thermistors--Thermistors would not be practical to plant in the lunar simulant due to the fact that they could change the drilling conditions or alter the environment, thereby affecting the outcome of the test results.

Temperature Strip--This strip, made up of reactive substances which change color at different temperatures, would also encounter the problem of having no line of sight. It would also be very difficult to obtain these reactive substances which could withstand the friction and wear forces produced by the bit and

the lunar simulant.

Radio-transmitted Signals--The noise generated by the motor, drilling procedure and vibrations would make a radio signal very difficult to be transmitted accurately to a receiver. Another problem with the radio signal is the transmission of the signal through the bit and the durability of the transmitter.

## CONCLUSIONS & RECOMMENDATIONS

Although we did not fulfill all of our objectives-- our design is not yet ready for fabrication and testing -- we feel that we have come a long way toward reaching our goal of developing effective lunar drilling technology. However, there is much work left to be done, and we feel that our work should be viewed as a point of departure for the further development of this technology rather than as a completed design.

### Bit/Auger

We feel that PDC drag bit technology may be used to drill lunar rock if we can find a way to effectively deal with the frictional heating of the cutting surfaces. We recommend that a detailed analysis be carried out on the use of a thermosyphon or heat pipe to remove heat from the cutters. Also, we feel that our auger will remove the cuttings from the hole if the formation is only moderately hard and the cutting dimensions are on the order of 0.5 mm. However, the fine cuttings generated when very hard rock is drilled may be difficult or impossible to move with a conventional auger. We recommend that the present auger be abandoned in favor of a stepped auger system which would be operated by linear reciprocating motion. This would require some method of moving cuttings from the bit face to the foot of the stepped auger.



We have not completed the design and analysis of the threaded drill string connections. Of course, any further development of this design project should include a detailed design of the connections including consideration of non-threaded connections.

The complex geometry and loading conditions of the bit/auger and the carbide studs required us to limit our analysis to highly simplified models which may be inappropriate. We recommend that finite element techniques be employed to obtain a stress analysis of the bit/auger, the studs, and the cutters.

Finally, tolerances and manufacturing techniques have not been specified. Obviously, before any of this design can be built, the tolerances and methods of manufacture must be specified.

#### Drill Drive Mechanism

There are several areas of the drill drive mechanism which need additional attention. A small pump is needed for the fluid vacuum seal. This pump should be mounted on either the motor platform or the drive platform with tubing leading to the four tube fittings of the seal. This will ensure a complete seal around the drill shaft.

Another area for improvement is the overall steel structure ( platforms, I-beams, and channels). These members have been

designed for large factors of safety, yet attention must be directed toward the welds and bolts used to attach these pieces. These components should be analyzed for stress and fracture, and their dimensions should be verified.

It is recommended that a 10 gage steel enclosure be constructed around the entire drive apparatus. Although dimensions have been specified, the structure is still in the preliminary design stage and needs much attention in the areas of welds, fasteners, and stress analysis. Also, a remote control panel should be located in an observation booth. The panel should have controls for both gearmotors, the fluid vacuum seal pump, and instrumentation to display pertinent drilling data.

## ACKNOWLEDGEMENTS

The Lunar Drilling Apparatus Design Group would like to acknowledge the following people for their assistance in designing or selecting equipment for this project.

- 1) Khawan Amwar, Allied Devices Corp.,  
Drawer E, Bladwin, N.Y. 11510, (516)-223-9100
- 2) Art Fisher, Standard Steel Specialty Co.,  
P.O. Box 20, Beaver Falls, Pa. 15010, (412)-846-7600
- 3) Richard Goldy, Thomson Saginaw Ball Screw Co., Inc.,  
P.O. Box 9550, Saginaw, Michigan 48608,  
(517)-776-4123
- 4) James G. Hartley, School of Mechanical Engineering,  
Georgia Institute of Technology
- 5) Anthony Koyenski, Jr. Key High Vacuum Products, Inc.,  
36 Southern Blvd., Nescanset, N.Y. 11767,  
(516)-360-3970
- 6) Curtis Wright, Duff Norton, P.O. Box 32605, Charlotte,  
N.C. 28232, (704)-588-0300
- 7) A. V. Larson, School of Mechanical Engineering, Georgia  
Institute of technology.

## REFERENCES

1. Glowka, D.A., Design Considerations for a Hard-Rock PDC Drill Bit, Sandia National Laboratories, Geothermal Technology Development Division 6241, 1985
2. Glowka, D.A., Implications of Thermal Wear Phenomena for PDC Bit Design and Operation, Sandia National Laboratories, Geothermal Technology Development Division 6241, 1985
3. Westinghouse Electric Corporation, Summary of Moderate Depth Lunar Drill Development Program From Its Conception to 1 July 1972, Final Report, Jun 1972
4. Maurer, W.C., Advanced Drilling Techniques, Chp. 22, pp. 587-589, Tulsa, Oklahoma, The Petroleum Publishing Company, 1980
5. Bornemann, A. et al., Sintered Carbide for Cutting Tools, Metals Handbook, 8th Ed., Vol. 3, 1967, pp. 316-322
6. Avallone, E.A. and Baumeiste, T. III, Marks' Standard Handbook for Mechanical Engineers, 9th Ed., New York, McGraw-Hill Book Company
7. Dunn, P.D. and Reay, D.A., Heat Pipes, 1st Ed., New York, Pergamon Press
8. Jacobs, M.H., Process and Engineering Benefits of Sputter-Iron-Plated Titanium Nitride Coatings, Surface and Coatings Technology, November 1986, Vol. 29, No. 3, pp. 221-237
9. Sollenberger, N.W., Performance and Evaluation of Titanium Nitride Coatings, Thesis, Georgia Institute of Technology, September, 1978
10. McGregor, K., The Drilling of Rock, CR Books Ltd, London, 1967.
11. McNair, Will L., The Electric Drilling Rig Handbook, Petroleum Publishing Co., Tulsa, Oklahoma, 1980.

Product Catalogs (consulted but not cited in text)

Bearing Technical Journal, FMC Corporation, Link-Belt Bearing Division, Indianapolis, Indiana, 1970.

Boston Gear, Boston Gear Incom International Inc., Quincy, Massachusetts, 1985.

Browning Power Transmission Equipment, Browning Manufacturing Division, Maysville, Kentucky, 1986.

Electra Motors, Dresser Industries, Inc., Anaheim, California, 1985.

Georgia Steel Supply Company, Atlanta, Ga. 1987.

Industrial Power Transmission Products, T.B. Wood's Sons Company, Chambersburg, Pennsylvania, 1985.

Link-Belt and Stearns Power Transmission Products, PTC Components, Inc., Indianapolis, Indiana, 1984.

## **APPENDICES**

## Appendix 1A

### Spherical Roller Bearing Calculations

For Drill Stem :

diameter = 1 11/16 in.

assume  $F_{a_{max}}$  (thrust load) = 3000 lb

$F_{r_{max}}$  (radial load) = 1300 lb

$n_{max}$  = 300 RPM

$$1) (C/P) = ((L \ n \ 60)/(1000000))^{0.3}$$

--for bearings with shaft diameter > 1 in.

where :

C = basic load rating

$L_{10}$  = rating life

n = speed

P = equivalent radial load

$$P = 0.67(F_r) + 2.25(F_a) \quad \text{if } (F_a/F_r) > 0.45$$

$$P = 0.67(1300) + 2.25(3000) \quad (F_a/F_r) = 2.3077$$

$$P = 7563.03 \text{ lb}$$

$(C/P)(P) = C$ , C = 18000 lb for bearings with 1 11/16 inner diameter.

$$-- 1800/7563.03 = (C/P) , \quad (C/P) = 2.38$$

$$2) L_{10} = ((C/P)^{10/3}(1000000)/(h \ 60))$$

$$L_{10} = ((2.38)^{10/3}(1000000)/(300 \ 60)) = 1000 \text{ hrs, which is acceptable.}$$

## Appendix 1B

### Calculations for Screw Jack Bearings

Shaft Diameter = 2.25 in.

Desired  $L_{10}$  min = 1000 hrs

$n_{\max}$  = 40 RPM

$$(C/P) = ((1000 \times 40 \times 60)/(1,000,000))^{0.3}$$

$$(C/P) = 1.3, \quad C = 23600 \text{ lb}$$

$$(C/P)(P) = C, \quad P = 23600/1.3 = 18153.85$$

$$P = 0.67(F_r) + 2.5(F_a) = 18153.85$$

--iteration yields

$$F_{a_{\max}} = 6000 \text{ lb}$$

$$F_{r_{\max}} = 4700 \text{ lb}$$



## Appendix 1C

### Ball Bearing Screw Jack Motor Calculations

To find the torque necessary to convert rotary motion to linear motion :

$T = 0.177(P)(L)$ , where  $P$  = load, and  $L$  = lead in inches

$T = 0.177(1500)(0.25) = 66$  in-lbs --for each screw jack

With a 0.25" lead, 4 revolutions will yield 1" of linear thread. Therefore, if a maximum penetration rate of 80 in/min is assumed, a speed of 320 RPM would be required.

Thus :

$\text{Required HP} = (n(2T))/(63000) = ((320)(2.66))/(63000)$

$\text{Required Hp} = 0.67$  HP

A 1.5 HP gearmotor will be used to insure a greater factor of safety and in the event greater thrusts are required.

## Appendix 1D

### Welded Bellows Calculations

Required travel = 8 ft. (approx. 2 meters)

8:3 ratio of maximum extended length to maximum compressed length.

$$a = 8 + b$$

$$b = (3/8)a$$

$$a = 8 + (3/8)a$$

$$a = 64/5 = 12.8 \text{ ft.}$$

$$b = (3/8)(12.8) = 4.8 \text{ ft}$$

Washers need approximately 1.5 inches difference in outer and inner diameter to obtain maximum flexibility

Since :

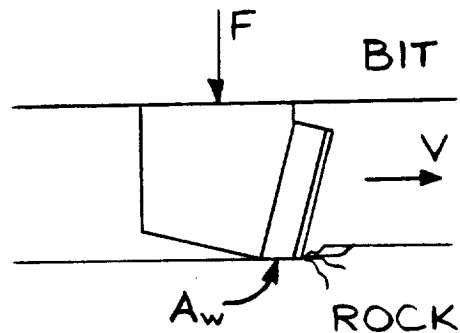
drill stem OD = 1 11/16 in.

Choose bellows ID = 2 in.

from this bellows OD = 3.5 in.

# APPENDIX 1E

## SINGLE-CUTTER DATA



$A_w$  = WEAR-FLAT AREA  
 $\sigma_c$  = COMPRESSIVE STRENGTH OF ROCK  
 $F$  = SINGLE-CUTTER THRESHOLD FORCE  
 $F_T$  = TOTAL THRESHOLD FORCE

FOR MODERATE WEAR ( $A_w = .02 \text{ in}^2$ ),  
PENETRATION BEGINS WHEN

$$F = \sigma_c A_w.$$

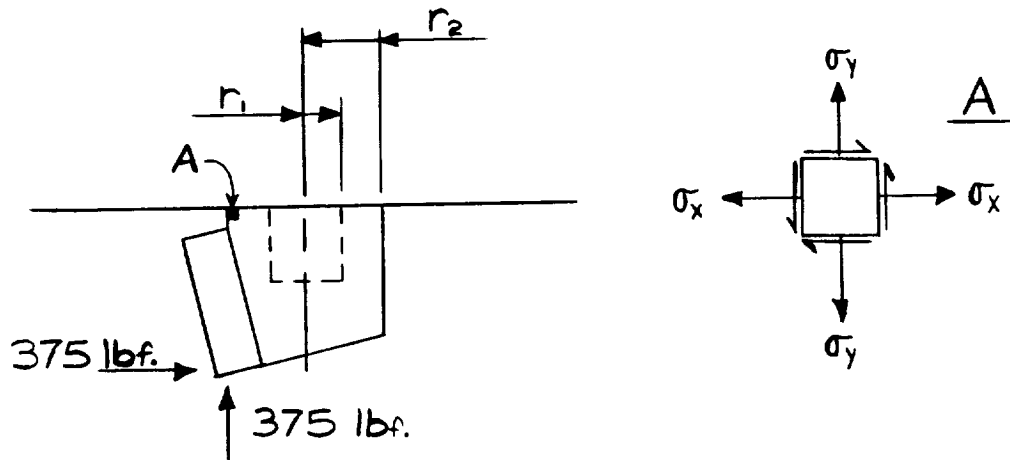
BECAUSE WE HAVE 8 CUTTERS,

$$F_T = 8F.$$

	$\sigma_c$ (kpsi)	$F$ (lbf.)	$F_T$ (lbf.)
DENSE BASALT	16.4	328	2624
HORNEBLENDE SCHIST	20.2	404	3232

# APPENDIX 1F

## CUTTER STRESS



$$\sigma_x = 0 = \tau_{xy} = \tau_{yx} \text{ (FREE SURFACE)}$$

$$\sigma_y = \left[ \frac{Mc}{I} + \left( \frac{-375 \text{ lbf.}}{A} \right) \right] K_T$$

$$M = (375 \text{ lbf.}) \left( \frac{8}{25.4} \text{ in.} \right) = 118.1 \text{ lbf.-in.}$$

$$c = r_2 = 4.1 \text{ mm} = 0.1614 \text{ in.}$$

$$r_1 = 2 \text{ mm} = 0.07874 \text{ in.}$$

$$K_T = \text{STRESS CONC.} = 3$$

$$I = \frac{\pi}{4} (r_2^4 - r_1^4) = (5.028)(10)^{-4} \text{ in.}^4$$

$$A = \pi (r_2^2 - r_1^2) = .06236 \text{ in.}^2$$

$$\sigma_y = \left[ \frac{(118.1 \text{ lbf.-in.})(.1614 \text{ in.})}{(5.028)(10)^{-4} \text{ in.}^4} - \frac{375 \text{ lbf.}}{.06236 \text{ in.}^2} \right] 3$$

$$= 95,694 \text{ psi}$$

$$\underline{\underline{\text{f.s.}}} = \frac{S_{UTS}}{\sigma_y} = \frac{170 \text{ Kpsi}}{95.7 \text{ Kpsi}} = \underline{\underline{1.78}}$$

# APPENDIX 1G

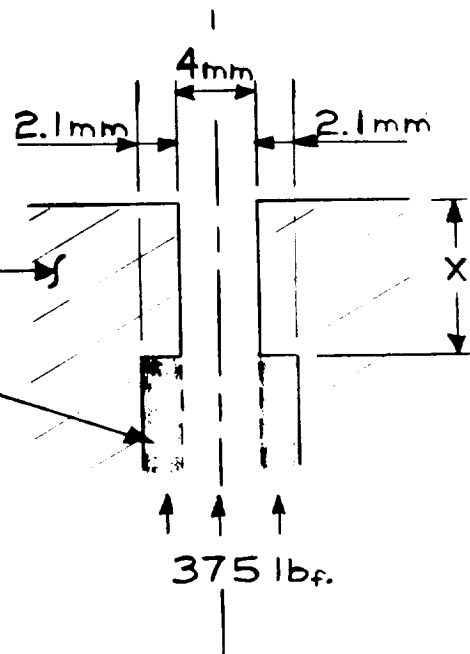
## BIT FACE THICKNESS

— DEFLECTION —

TREAT LIKE CANT. BEAM

BIT-BODY  
(4140)

STUD  
(WC)



$$\begin{aligned} l &= 2.1 \text{ mm} \\ b &= 2\pi (3.05 \text{ mm}) \\ x &= 8 \text{ mm} \end{aligned}$$

$$\omega = \text{UNIFORM LOAD} = \left( \frac{375 \text{ lbf.}}{2.1 \text{ mm}} \right) = 178.57 \frac{\text{lbf.}}{\text{mm}}$$

$$I = \frac{bh^3}{12} = 817.65 \text{ mm}^4$$

$$y_{\text{max}} = \frac{\omega l^4}{8EI} = \frac{(178.57 \frac{\text{lbf.}}{\text{mm}})(2.1 \text{ mm})^4}{(8)(30 \times 10^6 \text{ psi}) \left( \frac{1 \text{ in.}}{25.4 \text{ mm}} \right)^2 (817.65 \text{ mm}^4)}$$

$$\underline{y_{\text{max}} = .000011 \text{ mm}} \longrightarrow \text{OK}$$

— FAILURE BY SHEAR —

$$A = (8 \text{ mm})(2\pi)(3.05 \text{ mm}) = 153.3 \text{ mm}^2 = 0.2376 \text{ in.}^2$$

$$S_{\text{SY}} = S_{\text{UTS}} / 2 = 135 \text{ kpsi} / 2 = 67,500 \text{ psi}$$

$$\tau_{\text{MAX}} = \left( \frac{3V}{2A} \right) = \left[ \frac{3(375 \text{ lbf.})}{2(0.2376 \text{ in.}^2)} \right] = 2367.4 \text{ psi}$$

$$\underline{\underline{f.s. = \frac{S_{\text{SY}}}{\tau_{\text{MAX}}} = 28.5}}$$

# APPENDIX 1H

## BUCKLING OF DRILL STEM

EULER'S EQN. IS  $P_{CR} = \frac{n\pi^2 EI}{l^2}$

where

- $n$  = END COND. COEFFICIENT = 1  
 $E$  = YOUNG'S MODULUS =  $30(10)^6$  psi  
 $l$  = LENGTH = 48 in.  
 $I$  = MOMENT OF INERTIA =  $\pi(r_{OUT}^4 - r_{IN}^4)/4$   
 $r$  = PIPE RADIUS ( $r_{OUT} = 18$  mm)  
 $P$  = MAX LOAD = 3,000 lbs.  
 $f.s.$  = SAFETY FACTOR (TRY 3)  
 $P_{CR}$  = CRITICAL LOAD =  $P(f.s.) = 9,000$  lbs.

$$\frac{(4)(9,000 \text{ lbs.})(48 \text{ in.})^2}{\pi^3 (30)(10)^6 \text{ psi}} = \left[ \left( \frac{18}{25.4} \text{ in.} \right)^4 - r_{IN}^4 \right]$$

$$r_{IN} = 0.6354 \text{ in.} = 16.1 \text{ mm}$$

ANALYSIS DOES NOT ACCOUNT FOR HOLES IN PIPE,  $\Rightarrow$  USE  $r_{IN} = 12$  mm

$$P_{CR} = \frac{\pi^2 (30)(10)^6 \text{ psi} \left( \frac{\pi}{4} \right) \left[ \left( \frac{18}{25.4} \text{ in.} \right)^4 - \left( \frac{12}{25.4} \text{ in.} \right)^4 \right]}{(48 \text{ in.})^2}$$

$$P_{CR} = 20,427 \text{ lbs.} = f.s. (3,000 \text{ lbs.})$$

$$\underline{f.s. = 6.8}$$

# APPENDIX 1I

## TORQUE AND POWER REQUIREMENTS

$T_i$  = TORQUE CREATED BY CUTTER  $i$

$r_i$  = RADIAL LOCATION OF CUTTER  $i$

$F$  = THRUST PER CUTTER = 375 lbf.

$\mu_d$  = DRAG COEFFICIENT = 1

$F_d$  = DRAG FORCE PER CUTTER =  $\mu_d F = 375$  lbf.

$$T = \sum_i F_d r_i$$

$$T = 375 \text{ lbf.} [3(21.4 \text{ mm}) + 2(17.07 \text{ mm}) + 12.75 \text{ mm} + 8.92 \text{ mm} + 4.10 \text{ mm}]$$

$$T = 46,354 \text{ lbf.-mm} = 1825 \text{ lbf.-in.}$$

$$2T = 3650 \text{ lbf.-in.} = 304.17 \text{ lbf.-ft.}$$

$$200 \text{ rpm} = 20.94 \text{ rad./sec.}$$

$$P = \omega T = (304.17 \text{ lbf.-ft.})(20.94 \text{ rad./sec.})$$

$$\underline{P} = 6,369 \frac{\text{lbf.-ft.}}{\text{sec.}} = \underline{\underline{11.58 \text{ HP}}}$$

## Appendix 2

### Specifications of Drive Mechanism Parts

#### Fluid vacuum seal

Suggested Vendor : Key High Vacuum Products, Inc.

36 Southern Blvd.

Nesconset, N.Y. 11767

(516)-360-3970

Engineering Contact : Anthony Koyerski, Jr.

Part Number : 50D103328

Model Number : HDSB-1687-F-W-136

Maximum speed = 2870 RPM

Torque Capacity = 12,445 in-lb

Thrust Load Capacity = 5200 lb

Mounting Flange :

OD = 7 in.

ID = 5 in

Bolt circle = 6.0 in.

6 10-32 UNF bolts

"o"-ring size = 2-351

Material : Stainless Steel



Fluid Vacuum Seal Pump

Suggested Vendor : Key High Vacuum Products, Inc.

36 Southern Blvd.

Nesconset, N.Y. 11767

(516)-360-3970

Engineering Contact : Anthony Koyerski, Jr.

Part Number : N/A

\* 4 face mounting fittings

Ball Bearing Screw Jacks

Suggested Manufacturer : Thomson Saginaw Ball Screw Company,

Inc., P.O. Box 9550

Saginaw, Michigan 48608

(517)-776-4123

Engineering Contact : Richard Goldy

Ball Nuts

Part Number : 5703243

Nominal Size : 2.5 in.

Lead : 0.25 in.

\* Square

\* Right Hand Threads

Welded Bellows Seal

Suggested Vendor : Key High Vacuum Products, Inc.

36 Southern Blvd.

Nesconset, N.Y. 11767

(516)-360-3970

Engineering Contact : Anthony Koyerski, Jr.

Part Number : N/A

Maximum compressed height = 4.8 ft.

Maximum extended height = 12.8 ft.

Bellows inner diameter = 2 in.

Bellows outer diameter = 3.5 in.

Lower Flange :

OD = 32 in.

ID = 3.5 in.

Bolt Circle = 25 in.

14 1" - 8 UNF bolts

Upper Flange :

OD = 7 in.

ID = 2 in.

Bolt Circle = 6.0 in.

6 10 - 32 UNF bolts

"o"-ring size = 2-351

Spherical Roller Bearings and Flange Blocks

Suggested Vendor : Dixie Bearings

4990 South Fulton Industrial Blvd.

Atlanta, Ga.

(404)-691-7390

Suggested Manufacturer : Link-Belt Power Transmission Products

Screw-Jack Bearings and Flange Units

Bearing Size Number : B22436

Shaft Diameter = 2.25 in.

Flanged Unit Number : F-B22436H

- \* Cast Iron
- \* Spiral locking collar
- \* 4-bolt mounting (round)

Drill Stem Bearings and Flange Units

Bearing Size Number : B22527

Shaft Diameter = 1 11/16 in.

Flanged Cartridge Unit Number : CSE - B22527H

- \* Steel Housing
- \* Self-Aligning
- \* Two Spring Locking Collars

Screw-Jack Gearmotor

Suggested Vendor : Electra Motors

3620 Interstate 85

Suite 2115

Atlanta, Ga. 30340

(404)-458-7400

Suggested Manufacturer : Dresser Industries Inc.

Gearmotor Part Number : CP145TW-267AH

\* 1.5 HP

\* Singleworm Type B

\* 25:1 ratio

High Torque Drive Belts and Sprockets

Suggested Vendor : T.B. Woods Distribution Center

2933-C Amwiler Court

Atlanta, Ga. 30360

(404)-447-9090

Suggested Manufacturer : T.B. Woods Sons Company

440 N. Fifth Ave.

Chambersburg, Pennsylvania 17201

(717)-264-7161

Belts

\_\_\_\_ Part Number : P-64-8M-30-3280

\* 2 belts needed

Sprockets

Part Number : P-64-8M-30-SK

\* 8M pitch

Drill Drive Motor

Suggested Vendor : Motion Industries, Inc.

1810 Auger Drive

P.O. Box 1125

Tucker, Ga. 30085

(404)-939-6731

Suggested Manufacturer : Boston Gear

Incom International Inc.

14 Hayward Street

Quincy, Massachusetts 02171

1-800-343-3353

Catalog Number : RUTF-B

Item Code : 66274

\* 15 HP NEMA "C" face

### Reducer and Adapter

Suggested Vendor : Electra Motors  
3620 Interstate 85  
Suite 2115  
Atlanta, Ga. 30340  
(404)-458-7400

Suggested Manufacturer : Dresser Industries Inc.

Reducer Part Number : 500 RH

Adapter Part Number : 254 TC

- \* Singleworm NEMA "C" face

- \* 5:1 ratio

### Steel Plates, I-Beams, Channels and Angles

Suggested Vendor : Georgia Steel Supply Co.  
903 Huff Road N.W.  
Atlanta, Ga. 30377  
(404)-355-9510

### Steel Plates

Motor Platform : Hot Rolled Steel    Thickness = 0.5"  
Drive Platform : Hot Rolled Steel    Thickness = 1 "  
Test Platform : Hot Rolled Steel    Thickness = 2 "  
Enclosure Panels : Hot Rolled Steel    Thickness = 10 gauge

### I-Beams

Test Platform : 1) Standard 8" by 4" by 0.27"  
2) Wide Flange Beam 8" by 8" by 0.493" by 0.315"

### Channels

Test Platform 1) Standard 8" x 2.343" x 0.303"  
2) Standard 4" x 1.647" x 0.247"  
Drive Platform 1) Standard 5" x 1.885" x 0.325"  
2) Standard 3" x 1.41" x 0.17"  
Motor Platform 1) Standard 3" x 1.41" x 0.17"

### Angles

Motor Platform 1) Standard 1.5" x 1.5" x 0.25"  
2) Standard 4" x 4" x 0.5"

Appendix 3    RADIAL LOCATIONS OF CUTTERS AND STUD HOLES (mm)

	1	2	3	4	5
CUTTERS	4.10	8.42	12.75	17.07	21.4
HOLES	4.10	8.42	12.75	17.07	17.90



#### Appendix 4

#### Tolerances and Fits of Studs and Holes

Holes:     min   8.200

             max   8.215

Studs:     min   8.228

             max   8.237

Interference:   min   .016

                     max   .041

Fit:         ANSI FN4

# Appendix 5 COST ANALYSIS

QUANTITY	ITEM	UNIT COST	TOTAL \$
1	5 in Fluid Vacuum Seal	4362	4362
1	3.5 in Welded Bellows Seal	10000	10000
2	2.5 in Ball Nuts	50	100
2	2.5 in Screws	70	140
4	2.25 in Spherical Bearings	75	300
2	1 11/16 in Spherical Bearings	105	210
1	1.5 hp Gearmotor	469	469
2	HTD Belts	62	124
4	HTD Sprockets	80	320
1	NEMA Motor Reducer & Adaptor	4517	4517
1	Control Panel	4500	4500
17000	Steel	3.5	59500
1	KERNCO Microprocessor	375	375
1	Photo-Sensor	70	70
1	Torque Gage	4300	4300
1	Torque Gage	4410	8820
2	Load Cells	535	535
1	Display Pannel	295	295
1	Keptrol Controller	66	528
8	PDC Cutters	37	296
8	Carbide Studs		
Total cost (neglecting fabrication costs)			99761

**Appendix 6**

**Progress Reports**

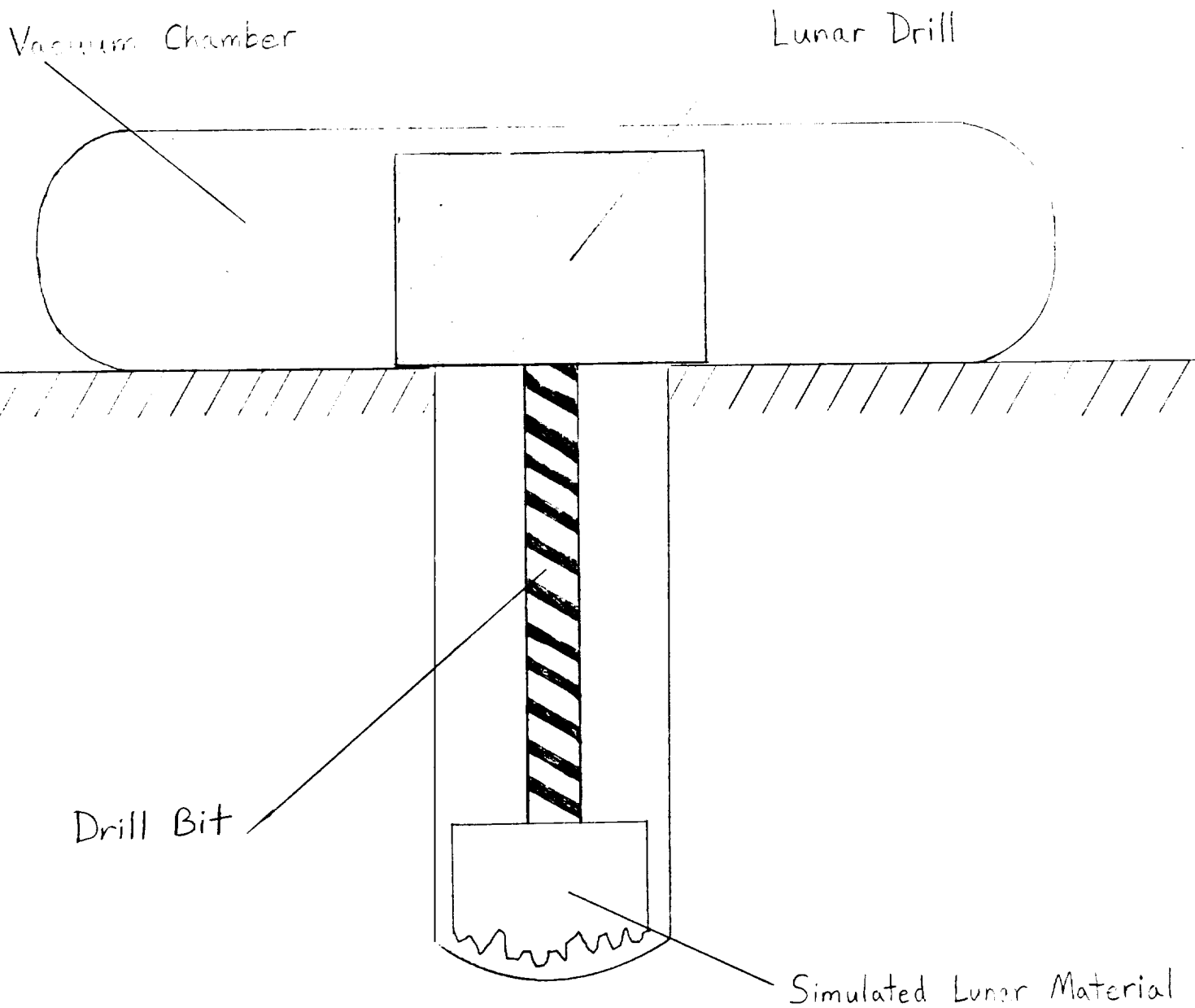
## PROGRESS REPORT

Drilling Apparatus for the Lunar Environment

After being presented with the need for developing a lunar drill, we began to gather information and develop a problem statement, and we discussed several conceptual alternatives. The resources we have so far consulted include reference books on lunar materials and heat pipes, the microfiche files in the Georgia Tech library, and Dr. J. G. Hartley, a professor of mechanical engineering at Georgia Tech who specializes in heat transfer. During the week, we discussed several alternatives for the design of the drill bit. We considered coring bits, rotating cone bits, and downhole hammers. We are currently in the information gathering stage of the design process, and any further decisions or developments will depend upon the gathering of more information, particularly information related to hand drilling operations that were conducted during the manned lunar missions of the late 1960's and early 1970's.

Group 7  
Oct. 4, 1988

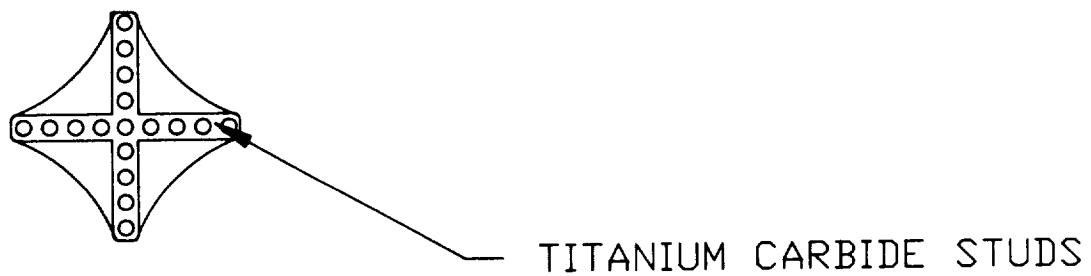
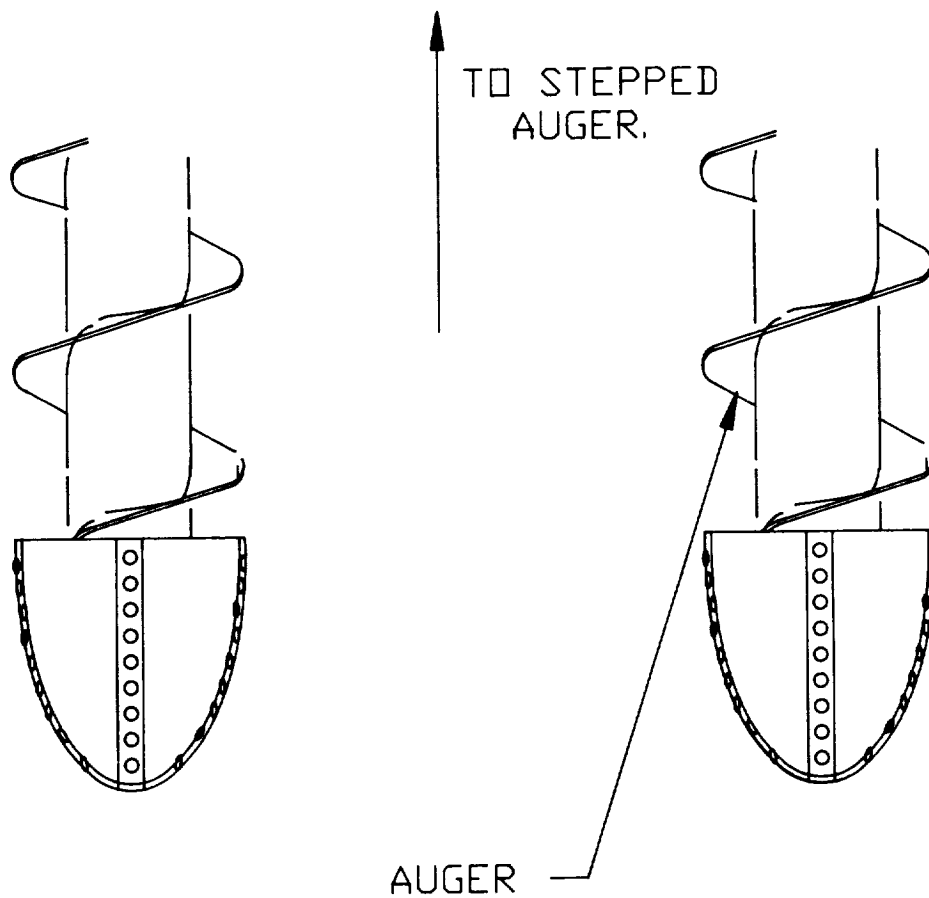
# Lunar Drilling Apparatus



**PROGRESS REPORT****Drilling Apparatus for the Lunar Environment**

During the past week we have continued to search for information related to our project, and we have had a productive brainstorming session. In our search for data we have found several NASA reports dealing specifically with lunar drilling and with the borehole jack that was used to drill core samples during the manned Apollo missions. As a result of our data search, we have decided that it is desirable to use mechanical methods to transport cuttings from the bit to the stepped auger removal system. Also, we have decided to consider a combination of rotary plus percussion action at the bit.

During a brainstorming session, we discussed several alternatives for bit design and cutting transport mechanism. One alternative which we think will work is shown in the figure. This is a bit for rotary plus percussion action with external fluting to move cuttings from the cutting face to the stepped auger system located up-hole.



POSSIBLE DRILL BIT CONFIGURATION

**PROGRESS REPORT**Lunar Drill and Test Apparatus

The progress of our group up until this week has been unsatisfactory. In an effort to change this, we have reorganized the design team into three working groups. The first group is responsible for the design of the drill bit, the second group is responsible for the design of the drive mechanism for the test facility, and the third group will select a lunar simulant and design the transducers and instrumentation for the test facility. The remainder of this report will address the progress of each working group separately.

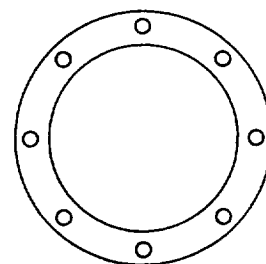
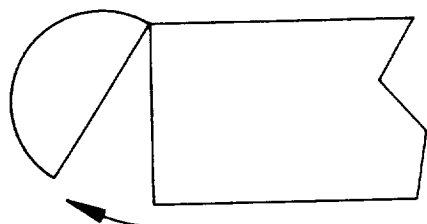
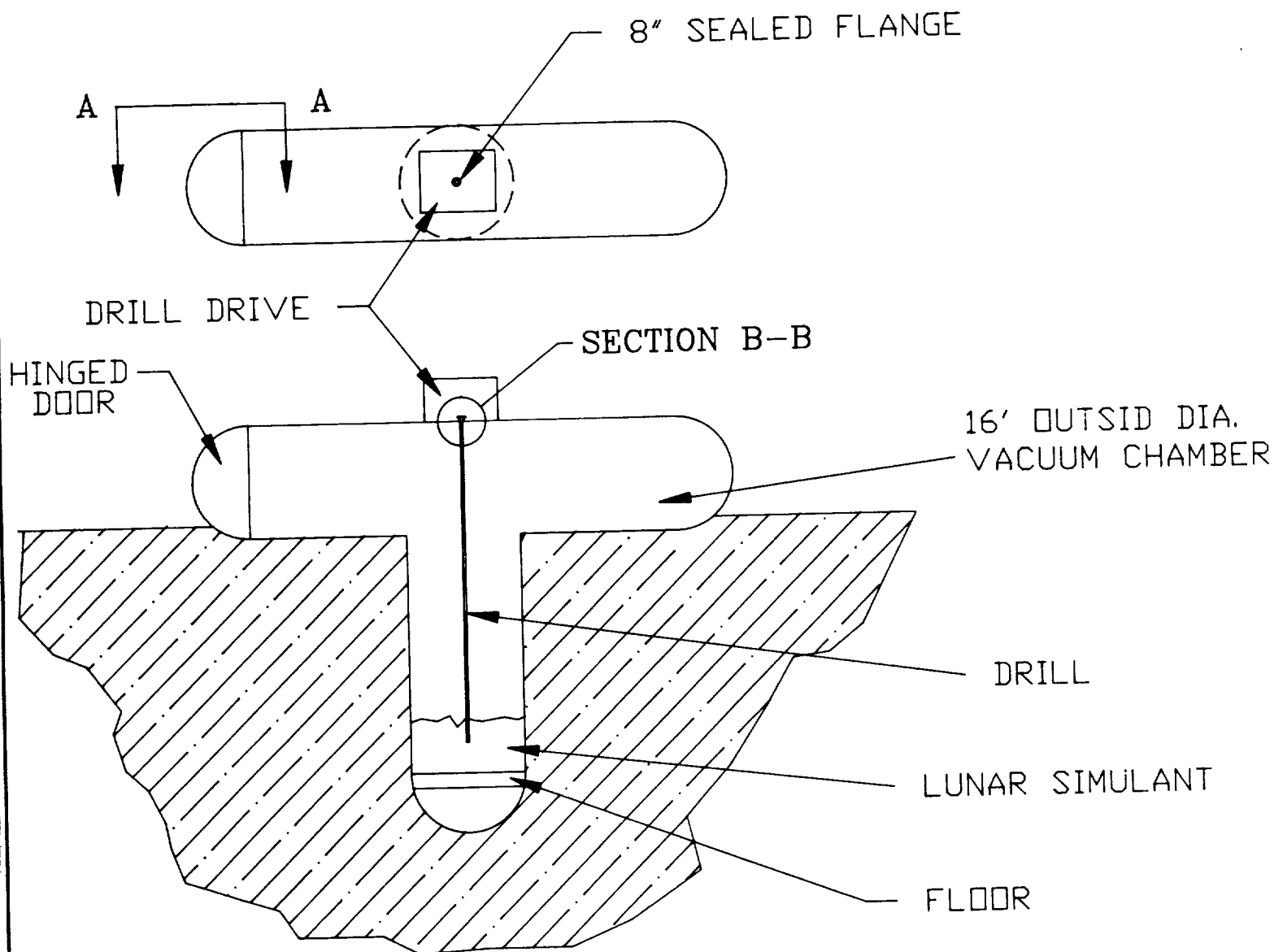
Bit Design We have decided to use a combination of rotary and percussive action to drill the bore-hole. The bit will be a modification of a conventional star bit. The modifications will include a half twist of the cutting face and flutes on the upper portion to auger cuttings to the initial stages of the stepped auger.

Drive Mechanism Design We have decided that the drive mechanism should be capable of rotating the drill at speeds varying from 0 to 500 RPM and of delivering up to 3000 hammer blows per minute (BPM). The percussive blows will be generated either by a pneumatic hammer or by an electrically driven hammer using a cam and spring. We have studied motor and hammer specifications from the Apollo 15 lunar surface drill (ALSD) to



get a rough idea of what is required.

Transducer/Instrumentation Design We have decided to use thermocouples for all downhole temperature measurements. We have investigated different thermocouple materials, and the final choice will depend upon an estimation of the temperature ranges that will be encountered. We have determined that rotational speed may be measured directly from the rotating machinery with existing "off the shelf" technology. We have begun to quantify the properties we require of a lunar simulant, but have not yet found any specific terrestrial mineral that will do the job.



# VACUUM TEST CHAMBER FOR LUNAR DRILL

## PROGRESS REPORT

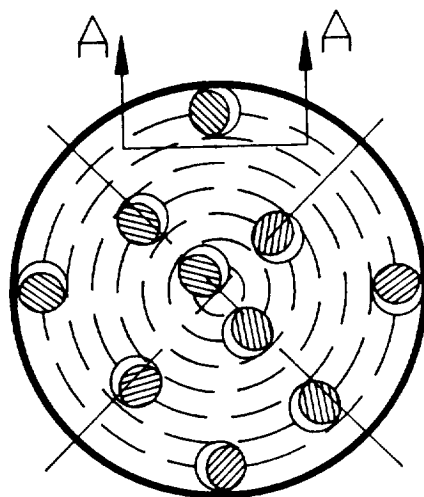
Lunar Drill and Test Apparatus

This week we are pleased to report that our group is beginning to make progress toward our design objectives, although we still feel that we are somewhat behind schedule.

Bit Design We investigated the feasibility of using a polycrystalline diamond compact rotary drag bit. Data from single cutter tests suggests that the penetrating forces required to cut very hard rock may be unattainable on the moon. We are now investigating recent information which accounts for the presence of nearby cutters in an actual drilling situation. We have considered several alternatives for the radial placement of the PDC cutters, and we have recently obtained criteria for optimum placement based on equal cutter wear. We have continued to investigate percussion bits since we are not yet convinced that the PDC drag bit will work.

Drive Mechanism Design This week we searched through vendor catalogs and found several pneumatic hammers which meet the requirements of our project. We also found several hand held rotary/percussion motors. We have selected a motor/reducer manufactured by Reliance which we will use to raise/lower the chip basket.

Transducer/Instrumentation Design We have identified basalt, vesicular basalt, schist, and dacite as possible lunar simulants based on a NASA report which indicates that these rocks have drilling properties similar to those of lunar rock.



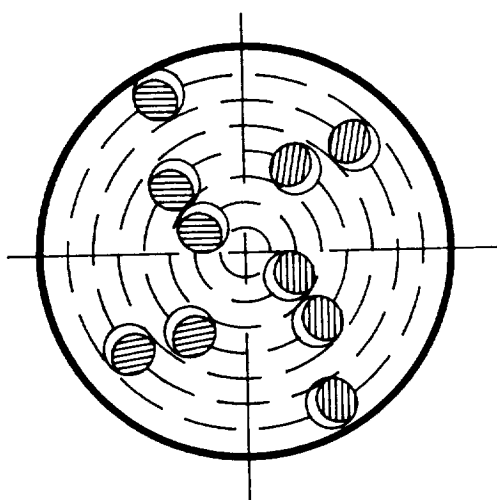
VERSION 1

PDC CONTACT  
SURFACE

STUD



SECTION A-A  
PDC STUD



VERSION 2

ALTERNATIVE RADIAL LOCATIONS OF  
POLYCRYSTALLINE DIAMOND COMPACT (PDC) CUTTER BITS

10/25/88

## PROGRESS REPORT

## LUNAR DRILL AND TEST APPARATUS

## DRILL AND BIT DESIGN

The time this week was concentrated on optimizing the placement of the PVC cutters on the drill bit. The major difficulty encountered is that the formulas used for these calculations are experimental correlations. As a result, they require experimentally determined coefficients and exponents. Unfortunately, these values are not readily available. However, the criterion for equal cutter wear can be approximated by increasing cutter density with increasing radial location. The present objective involves a design based upon this theory.

## TRANSDUCERS AND INSTRUMENTATION

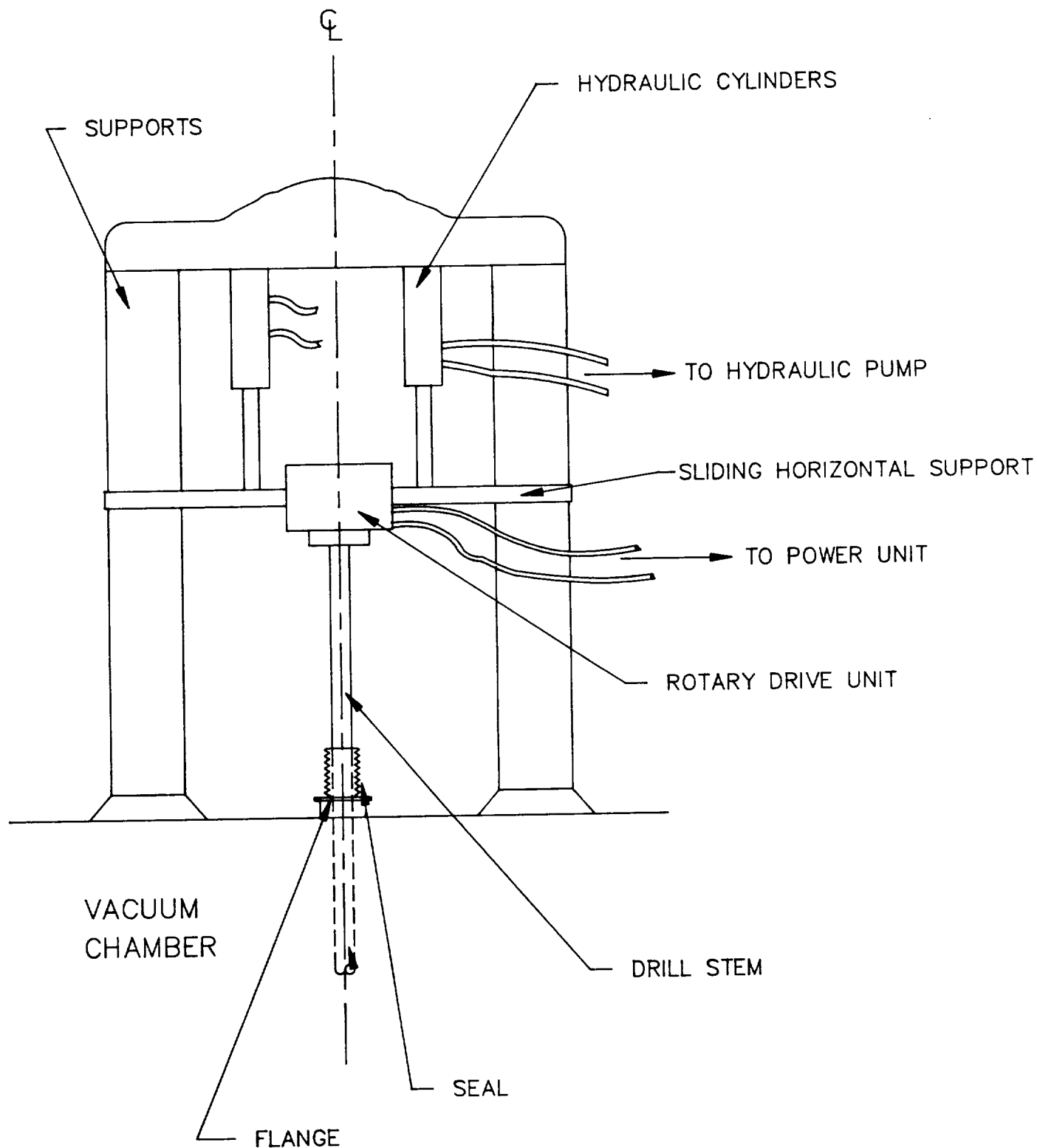
This week it was determined that the drill bit weight and torque may be evaluated indirectly from pressure measurements. This theory is particularly important if hydraulics are used to power the drilling apparatus. It has been decided that RPM will be measured directly from the rotary device. Also, we investigated methods of measuring the rate of penetration by incorporating commercially available equipment.

## DRILL DRIVE SYSTEM

The following basic system was agreed upon this week. (see figure) In addition to the support structure, there are:

- 1) Two hydraulic cylinders, controlled by either a piston driven or gear driven pump, to raise and lower the drill.
- 2) A rotary drive unit which will either be a variable-speed, reversible, electric motor or a hydraulic power unit utilizing the same fluid as the cylinders.

Also, the system will have a flexible accordion seal at the flange of the vacuum chamber. The entire structure will be mounted on a platform capable of supporting the 2,500lb force created by the drilling.



DRILL APPARATUS FOR TEST SITE

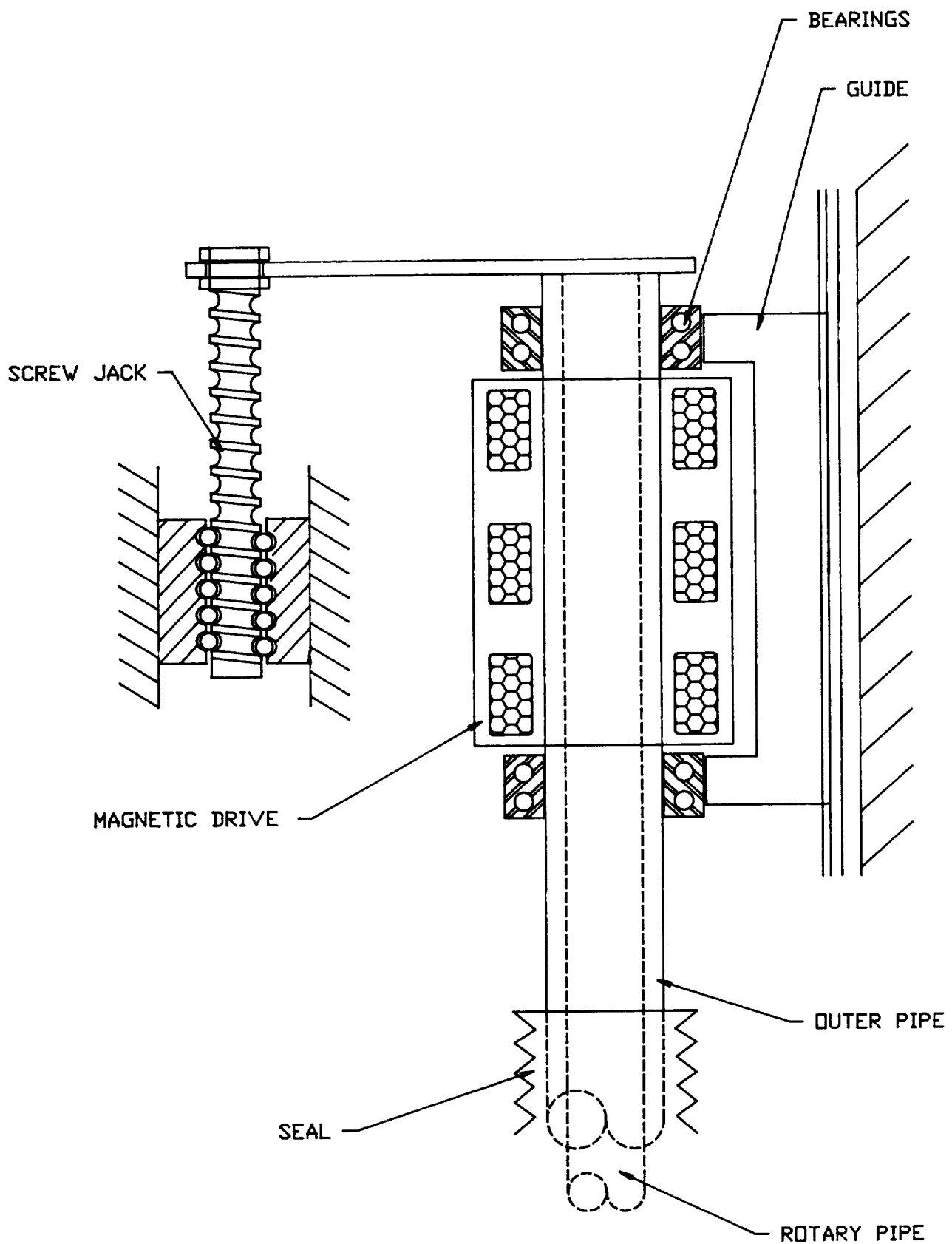
## PROGRESS REPORT

Lunar Drill and Test Apparatus

Bit Design We are currently working on the final design of the PDC drill bit. The bit will utilize eight 8.2 mm dia cutters mounted on Wc-Co studs. The cutters will be radially located so as to approximate equal force per cutter conditions. Cuttings will be moved to the outside of the bit face by mechanical action where they will travel upward through three vertical channels which feed into three auger flights.

Transducers/Instrumentation This week we investigated the feasibility of using infrared temperature sensors to measure downhole temperatures. We also evaluated several programable, digital readout thermocouple thermometers as well as several optical and mechanical RPM transducers. Based on previous lunar drill design studies, we have decided upon two possible lunar simulants and are currently designing the actual geometry of the specimen and a process for cooling it to lunar temperatures.

Drive Mechanism Design This week we have redesigned the drive unit to utilize a vertical rack and pinion or a ball bearing screw jack to provide vertical motion and thrust. The jack ( or rack) will be mounted in parallel with the drill shaft to reduce overall height. We are investigating the feasibility of using magnetic coils to provide rotational motion. This will allow the rotating shaft to be inclosed by a non-rotating sleeve and eliminate the problems associated with maintaining a vacuum seal against a shaft which rotates and moves vertically.



FOLDED DRILL DRIVE APPARATUS



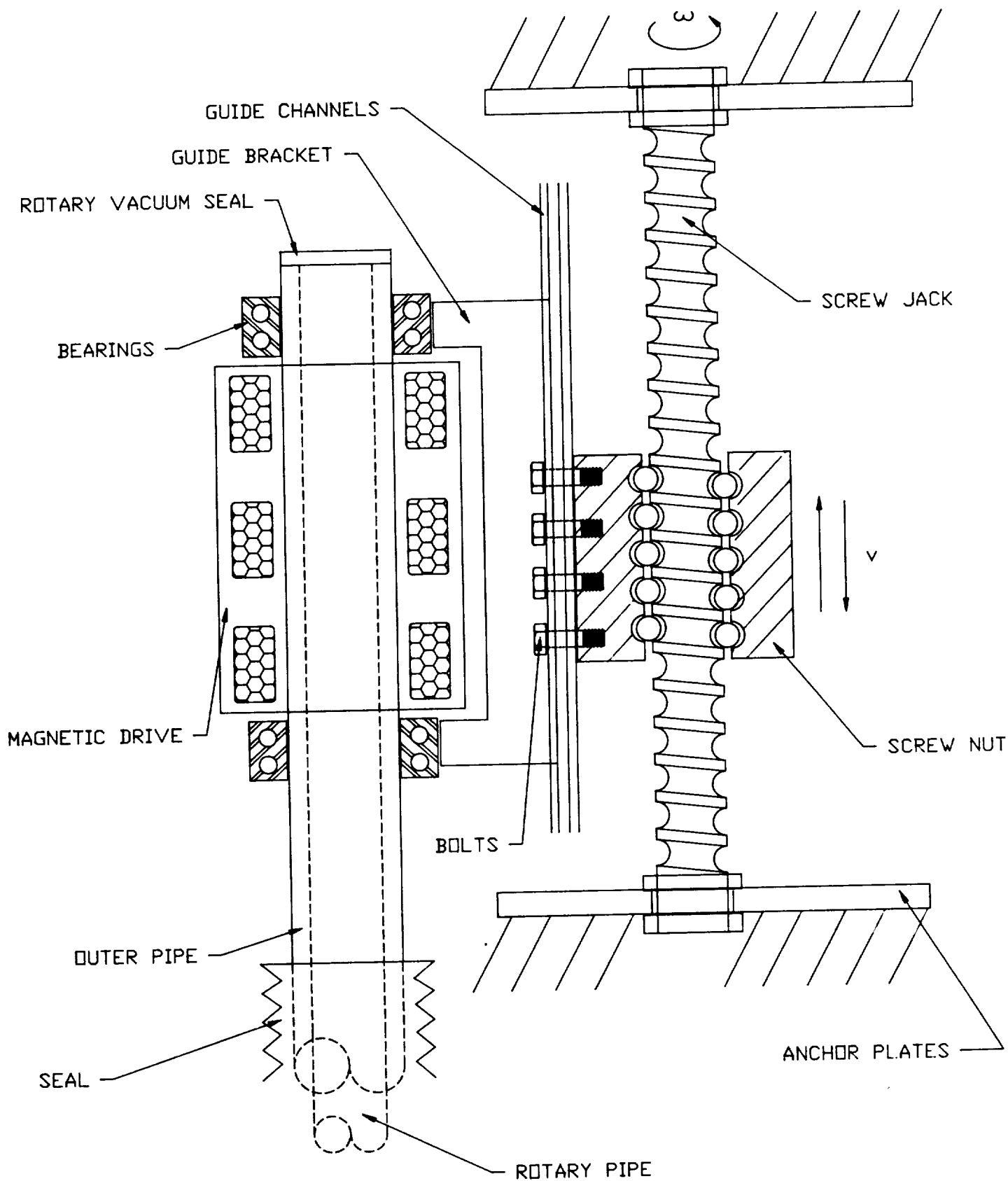
## PROGRESS REPORT

Lunar Drill and Test Apparatus

Bit Design - This week we have begun to finalize our bit design. We will use eight PDC cutters located radially so that the wear on each cutter will be approximately equal. We will use a three flight auger with a rake angle of 20.5 degrees to move the cuttings from the bit to the first stage of the stepped auger. We will incorporate a heat pipe into the drill stem to remove excess heat from the bit. We are presently working on a stress analysis of the PDC cutter mounting studs and the design of the heat pipe. We have determined the torque, thrust, and power required to drive the bit.

Drill Drive Mechanism - In our search for a linear synchronous motor, we found that the available motors range from 250 hp to 1800 hp. These values were reached between 500 RPM and 1500 RPM. These motors are obviously larger than that required by our system. We intend to telephone suppliers to determine if a motor can be manufactured to meet our specifications. In addition, we have made modifications to the overall drive structure to reduce weight and complexity.

Lunar Simulant - Three lunar simulants were chosen this week for different testing situations, they are basalt, gneiss, and schist. The three simulants can be separated according to their drillability and hardness. These rocks were specified as good simulants of lunar subsurface rocks in the NASA report "The feasibility study of a moderate depth lunar drill". The simulant will be cut into a block of dimensions 8 ft by 4 ft by 4 ft and will weigh between 2,100 and 2,500 pounds depending on the simulant.



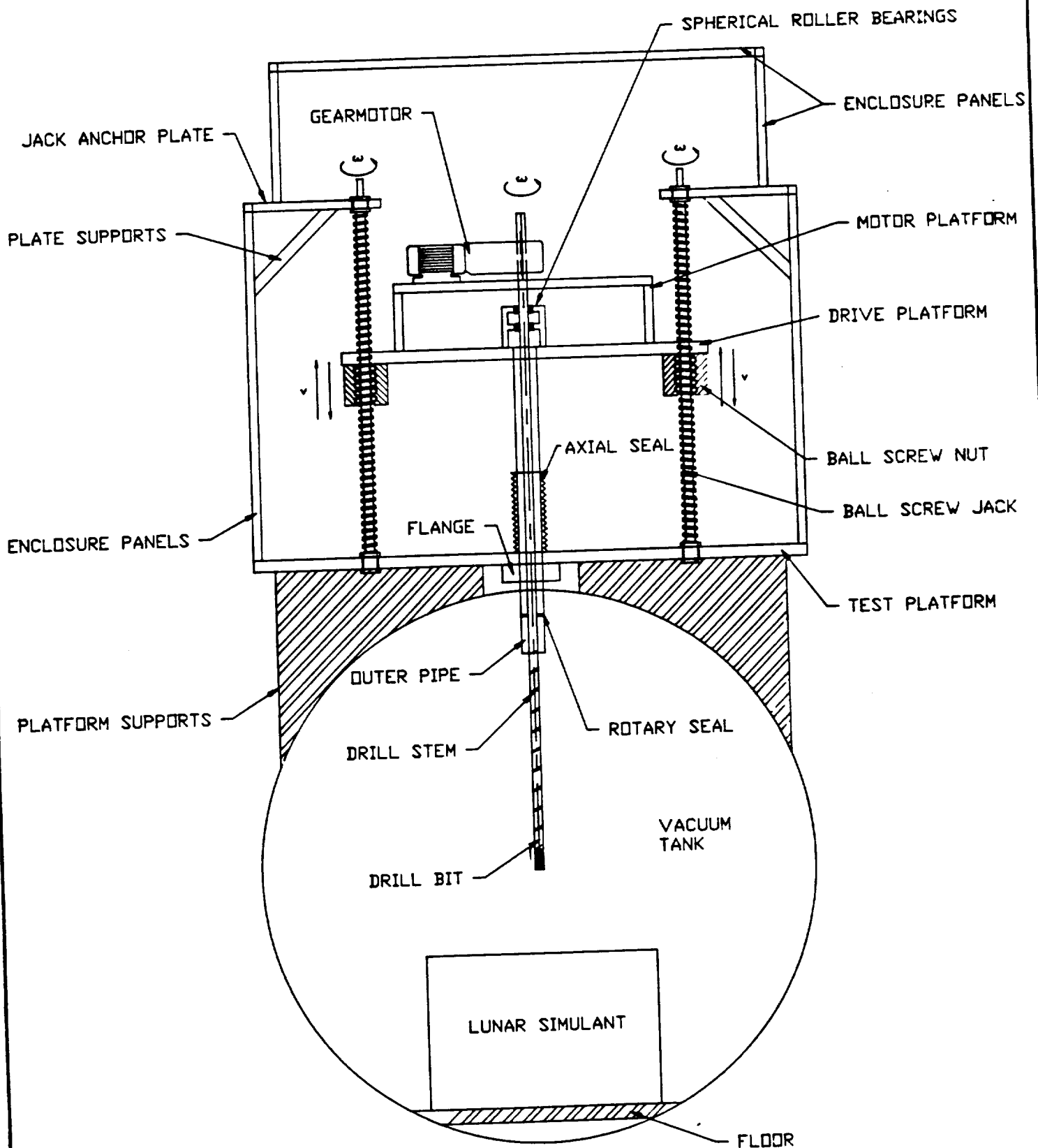
SIMPLIFIED DRILL DRIVE UNIT

**PROGRESS REPORT****Lunar Drill and Test Apparatus**

**Bit Design** We have completed the design of the bit and are presently working on the drawings. The bit material will be 4140 steel. We are considering eliminating the stepped auger from the design since this will result in a simpler, more predictable design which should accomplish the same tasks.

**Drive Mechanism Design** In the past week, we have begun to finalize the design of the drive system. We have located vacuum seals for the flange connection, and we have selected the motor and reducer. We have selected thrust bearings to support and apply thrust to the drill stem which meet the maximum thrust requirement of 3000 lb<sub>f</sub>. We have designed the supporting platform and the enclosure, but this design has not been finalized.

**Transducers/Instrumentation** We have selected an optical RPM sensor which utilizes infrared light. We will measure torque either directly from the motor, or indirectly as a function of the electric current driving the motor. We will obtain thrust measurements directly from the ball bearing screwjacks.

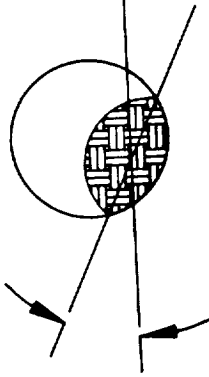


DRILL PLATFORM UNIT

## **Appendix 7**

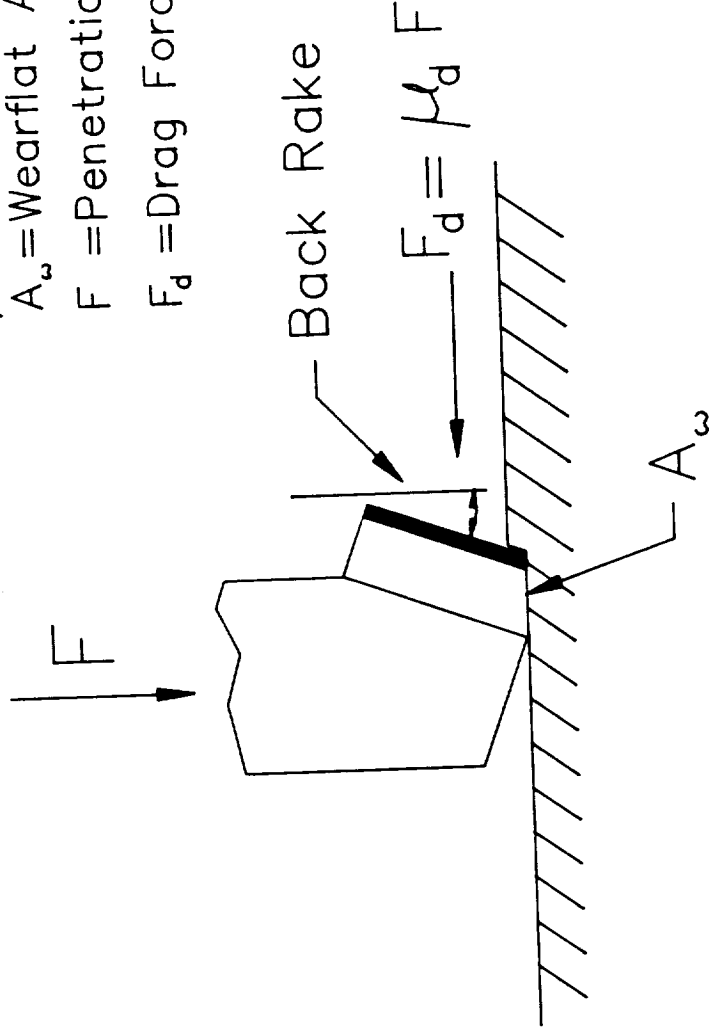
### **Figures**

Side Rake



Bit Center

$\mu_d$  = Coefficient of Drag  
 $A_w$  = Wearflat Area  
 $F$  = Penetration Force  
 $F_d$  = Drag Force



GEORGIA INSTITUTE OF TECHNOLOGY  
 ME 4182 DESIGN GROUP 7 - LUNAR DRILL

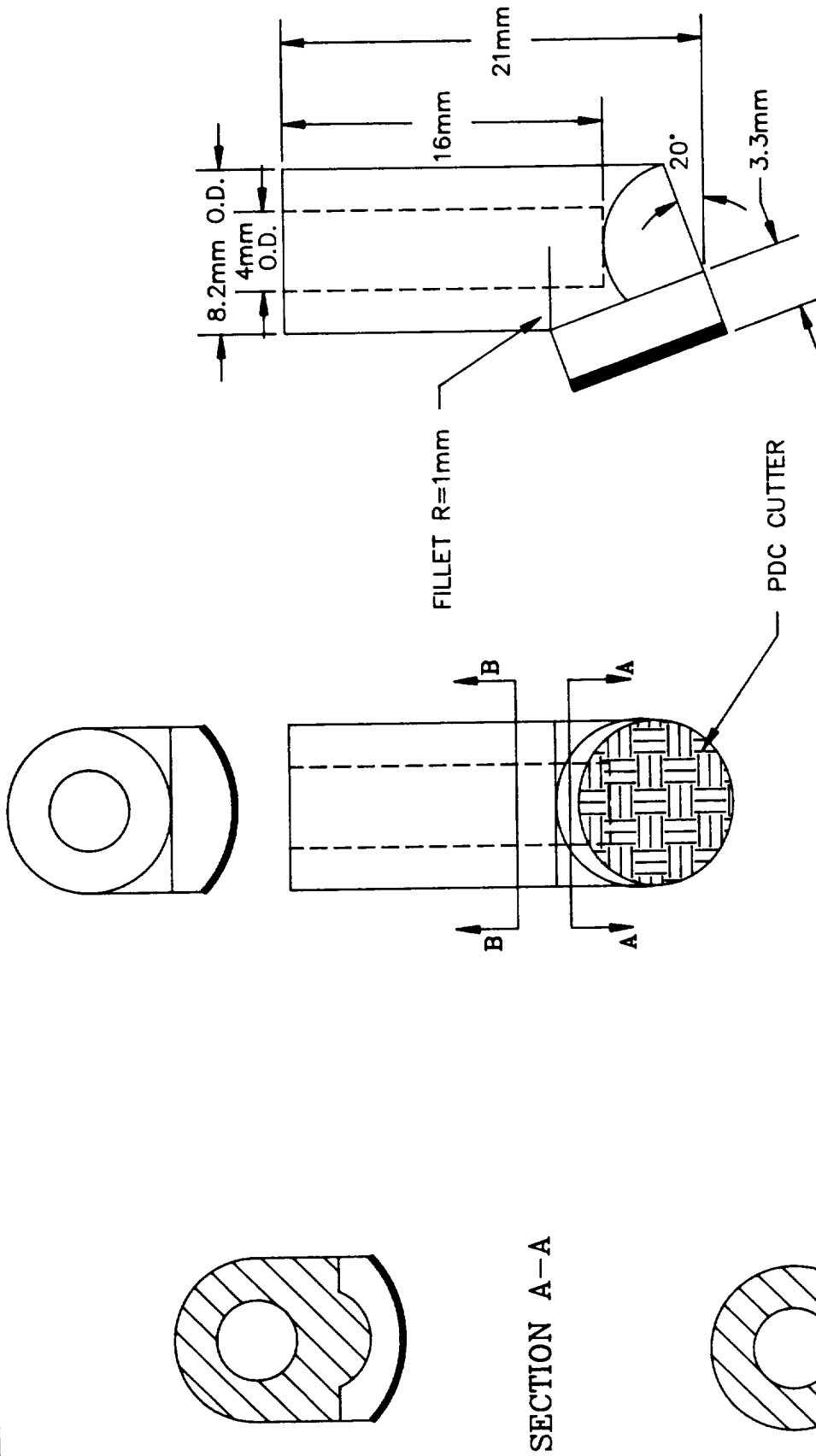
SCALE: 1/4" = 1/4" FIGURE # 1.1

DATE: 27 NOV 88

FNV

PDC

TERMINOLOGY

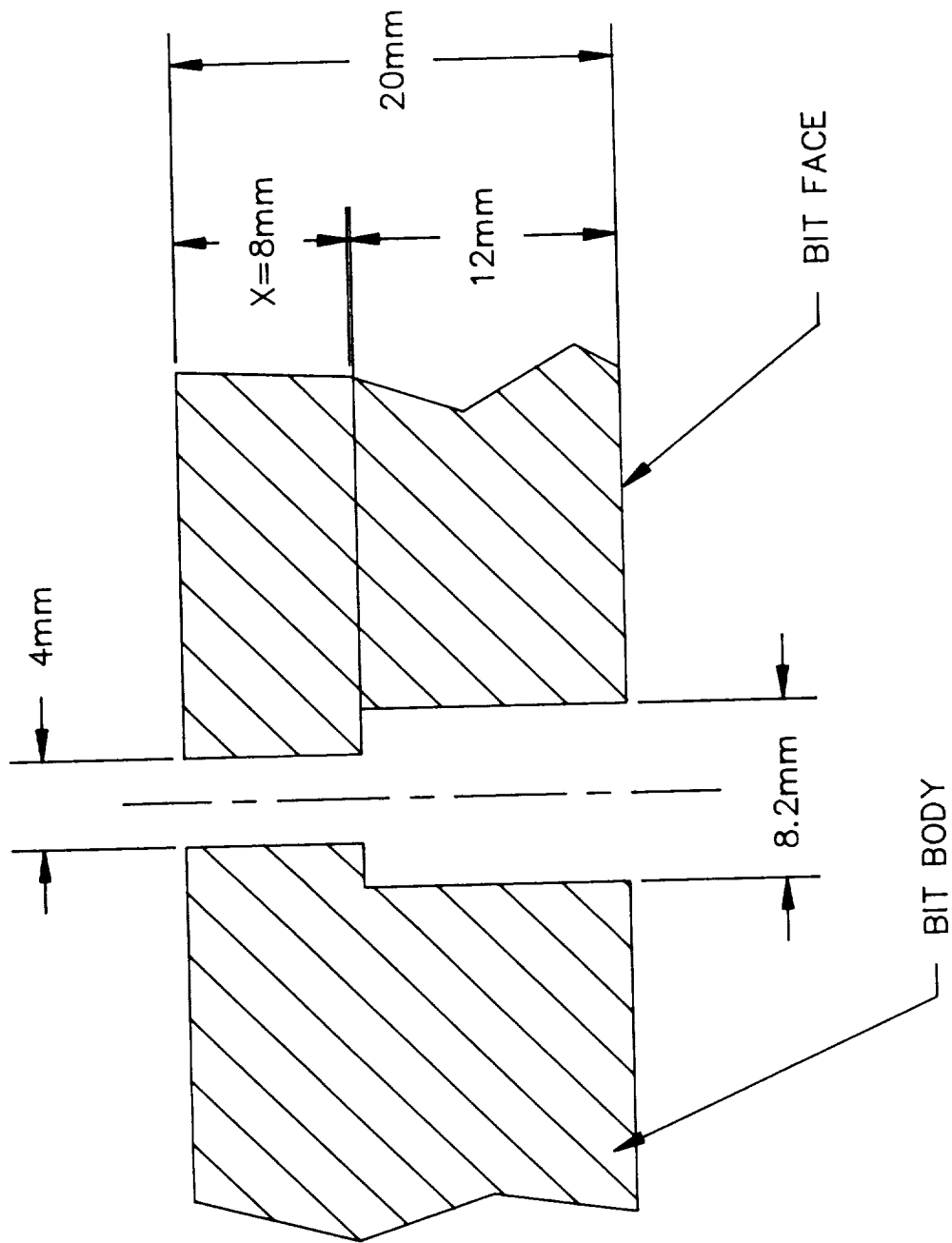


GEORGIA INSTITUTE OF TECHNOLOGY  
ME 4182 DESIGN GROUP 7 - LUNAR DRILL

SCALE: 1"=1/4" FIGURE # 1.2

DATE: 27 NOV 88 FNV

PDC CUTTERS WITH  
CARBIDE STUDS



GEORGIA INSTITUTE OF TECHNOLOGY  
ME 4182 DESIGN GROUP 7 - LUNAR DRILL

SCALE: 1/2"=1"

FIGURE #

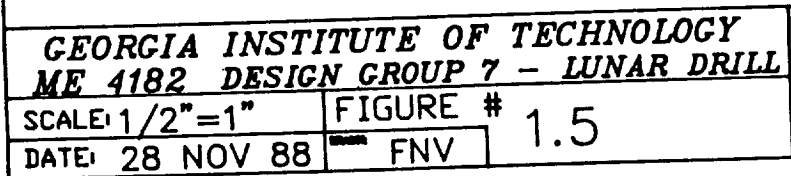
1.4

DATE: 28 NOV 88

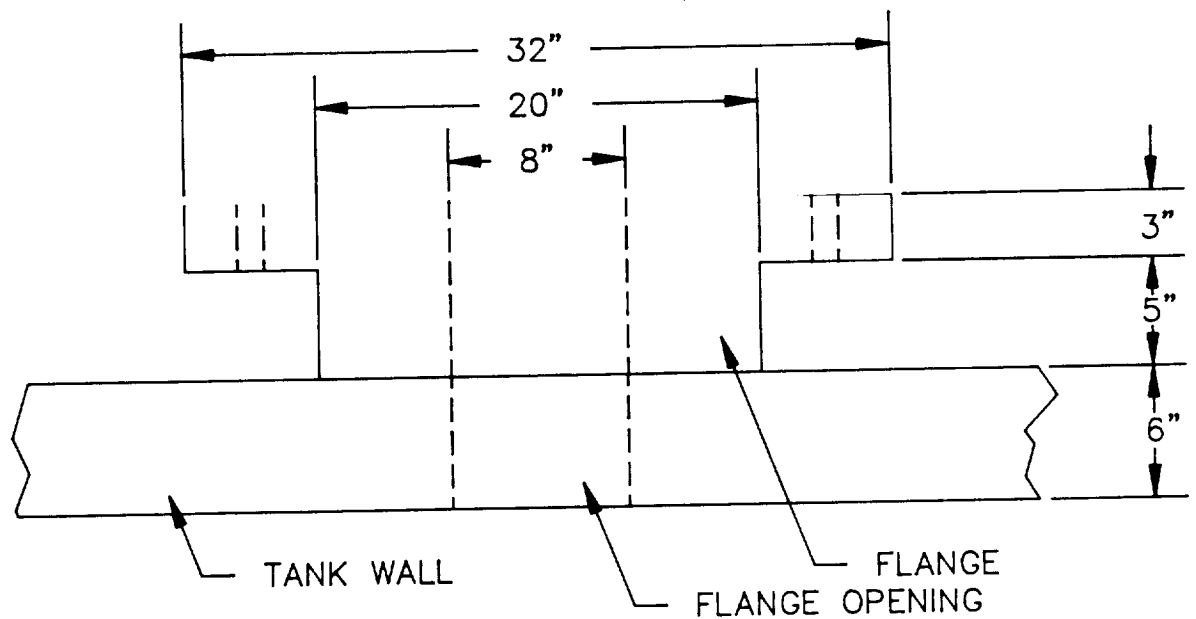
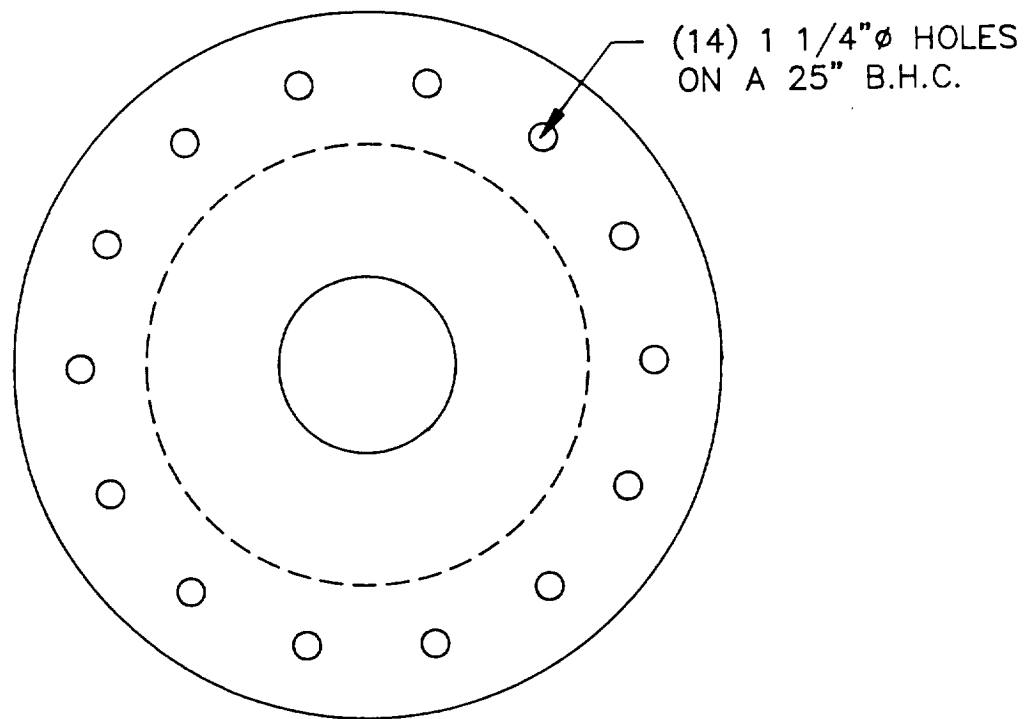
FNV

STUD HOLE  
CROSS SECTION





BIT/AUGER  
CROSS SECTION



GEORGIA INSTITUTE OF TECHNOLOGY  
ME 4182 DESIGN GROUP 7 - LUNAR DRILL

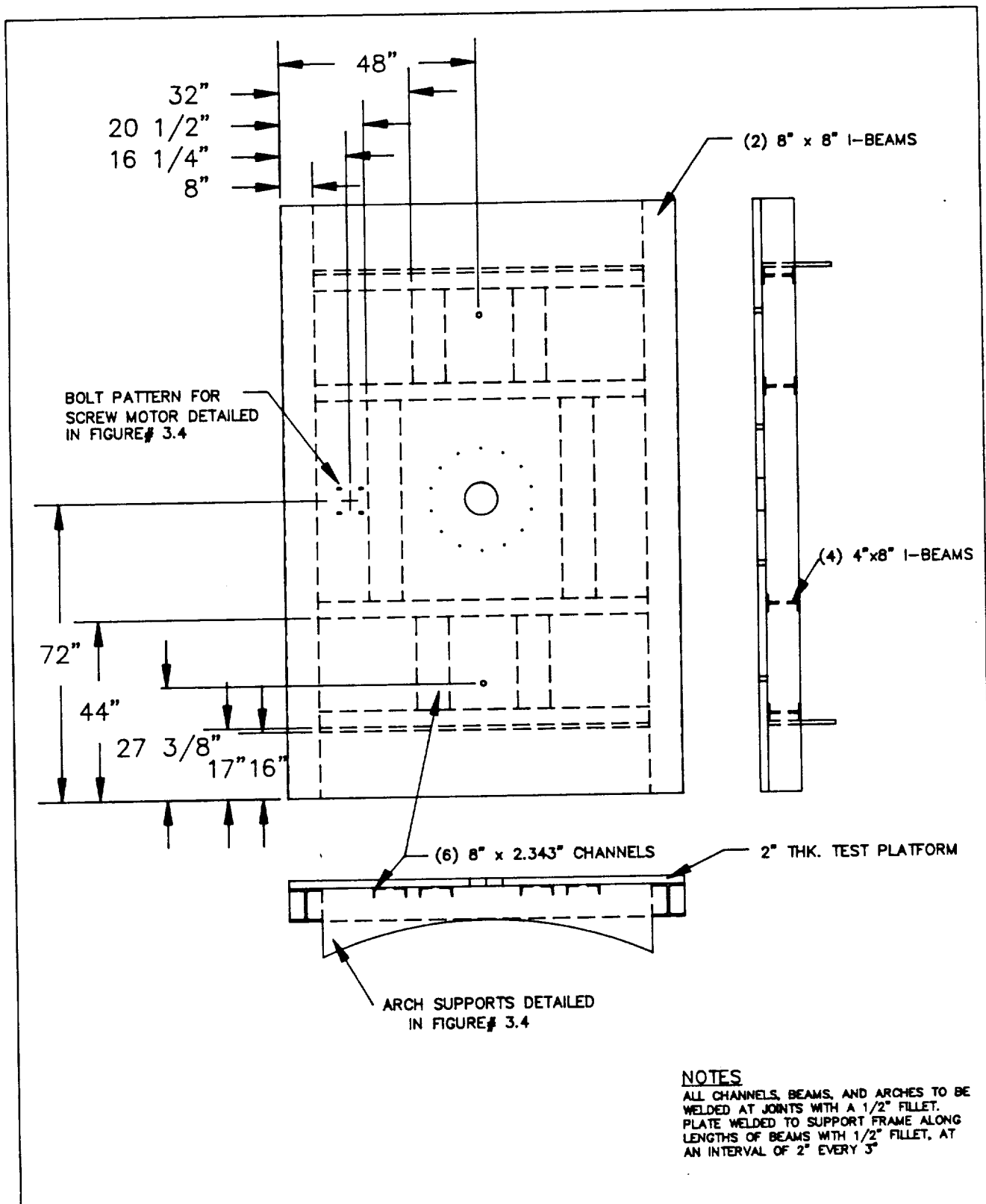
SCALE: 1/8"=1"

FIGURE # 3.2

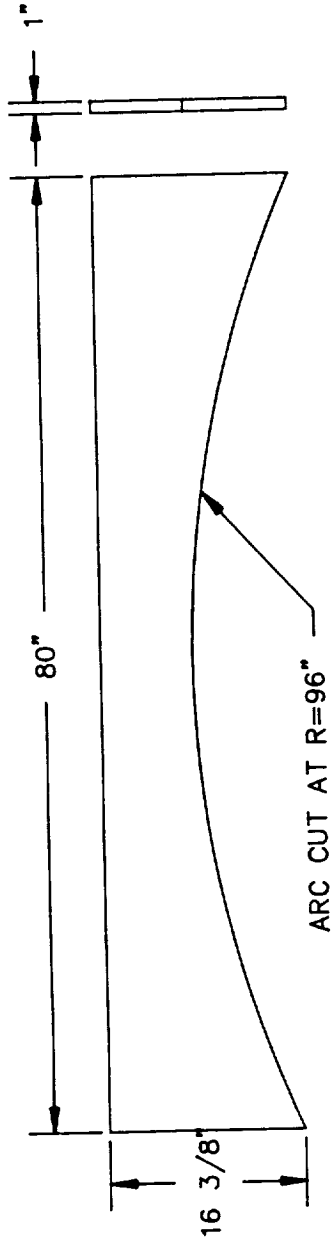
DATE: 25 NOV 88

FNV

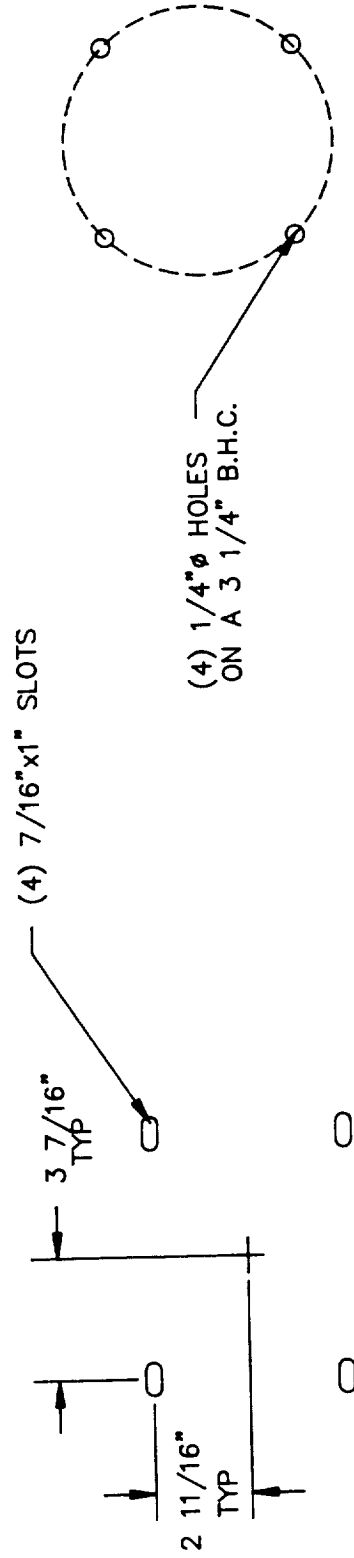
VACUUM TANK  
FLANGE



GEORGIA INSTITUTE OF TECHNOLOGY ME 4182 DESIGN GROUP 7 - LUNAR DRILL			TEST PLATFORM ASSEMBLY
SCALE: 3/8"=1'	FIGURE # 3.3		
DATE: 25 NOV 88	DESIGNER: FNV		



TEST PLATFORM ARCH SUPPORTS

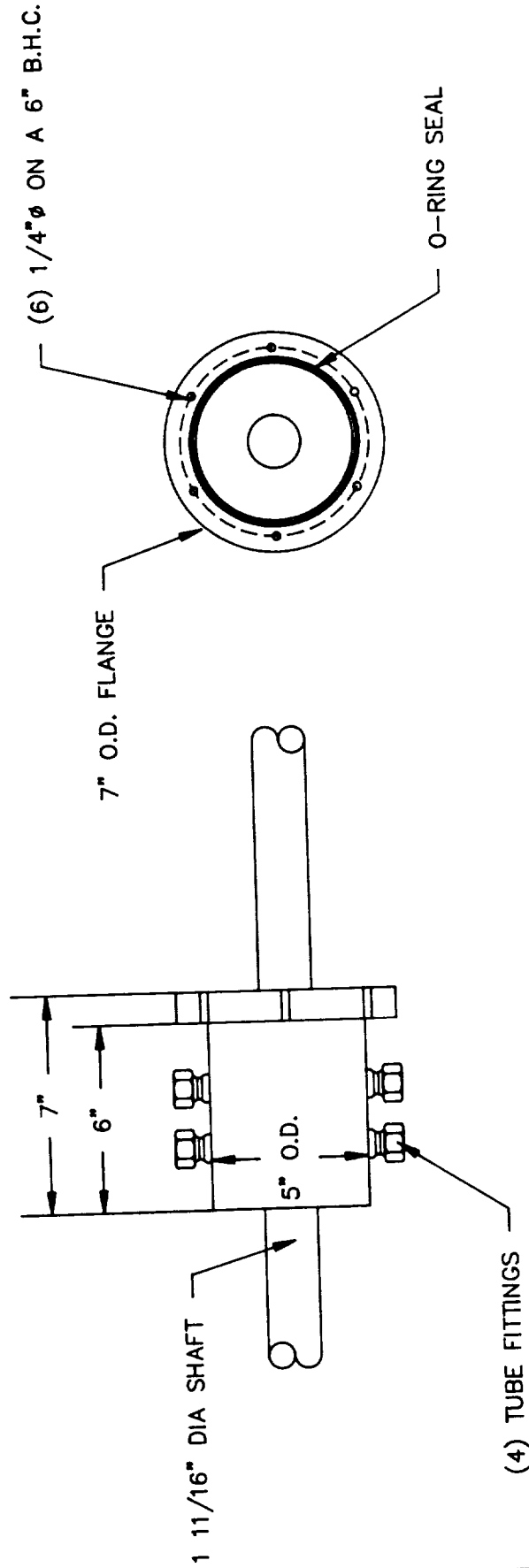


THRUST BEARING BOLT PATTERN

SCREW JACK MOTOR BOLT PATTERN

GEORGIA INSTITUTE OF TECHNOLOGY		
ME 4182 DESIGN GROUP 7 - LUNAR DRILL		
SCALE: 1"=1'	FIGURE #	3.4
DATE: 27 NOV 88	DESIGNER	FNV

ARCH SUPPORTS &  
BOLT DETAILS FOR  
TEST PLATFORM



# ROTARY VACUUM SEAL

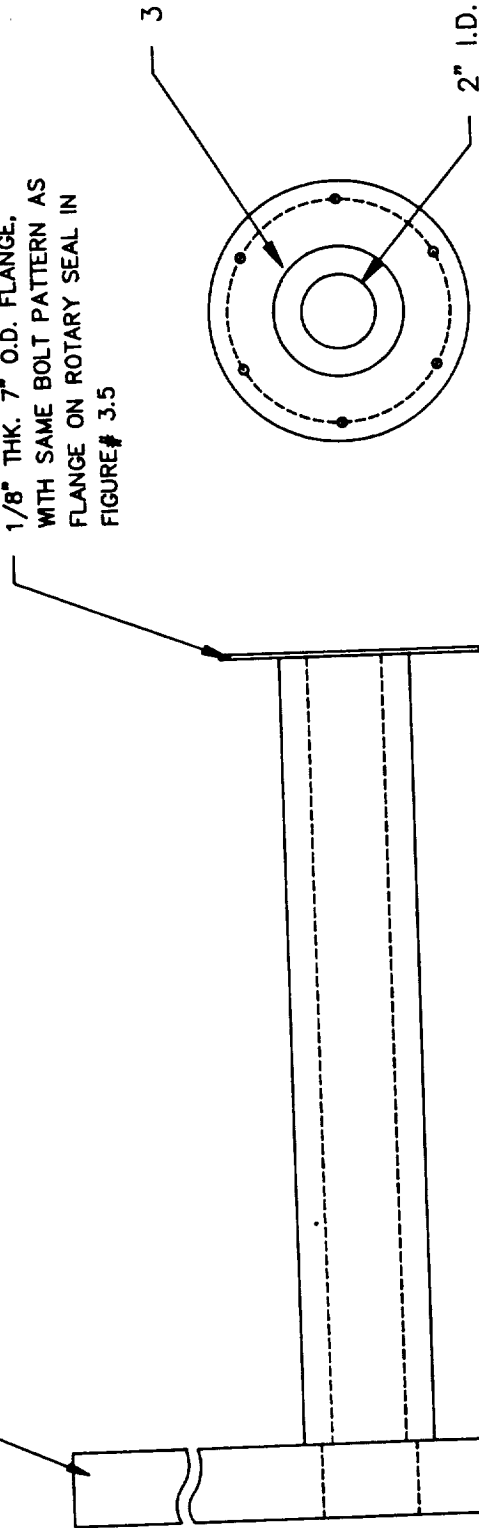
GEORGIA INSTITUTE OF TECHNOLOGY			
ME 4182 DESIGN GROUP 7 - LUNAR DRILL			
SCALE: $1/4"=1"$	FIGURE #		3.5
DATE: 27 NOV 88	FNV		

32" O.D. FLANGE IS 2" THK. AND HAS THE SAME BOLT PATTERN AS VACUUM TANK FLANGE IN FIGURE# 3.2

1/8" THK. 7" O.D. FLANGE, WITH SAME BOLT PATTERN AS FLANGE ON ROTARY SEAL IN FIGURE# 3.5

3 1/2" O.D. SHAFT

2" I.D.



GEORGIA INSTITUTE OF TECHNOLOGY

ME 4182 DESIGN GROUP 7 - LUNAR DRILL

SCALE: 1/4"=1"

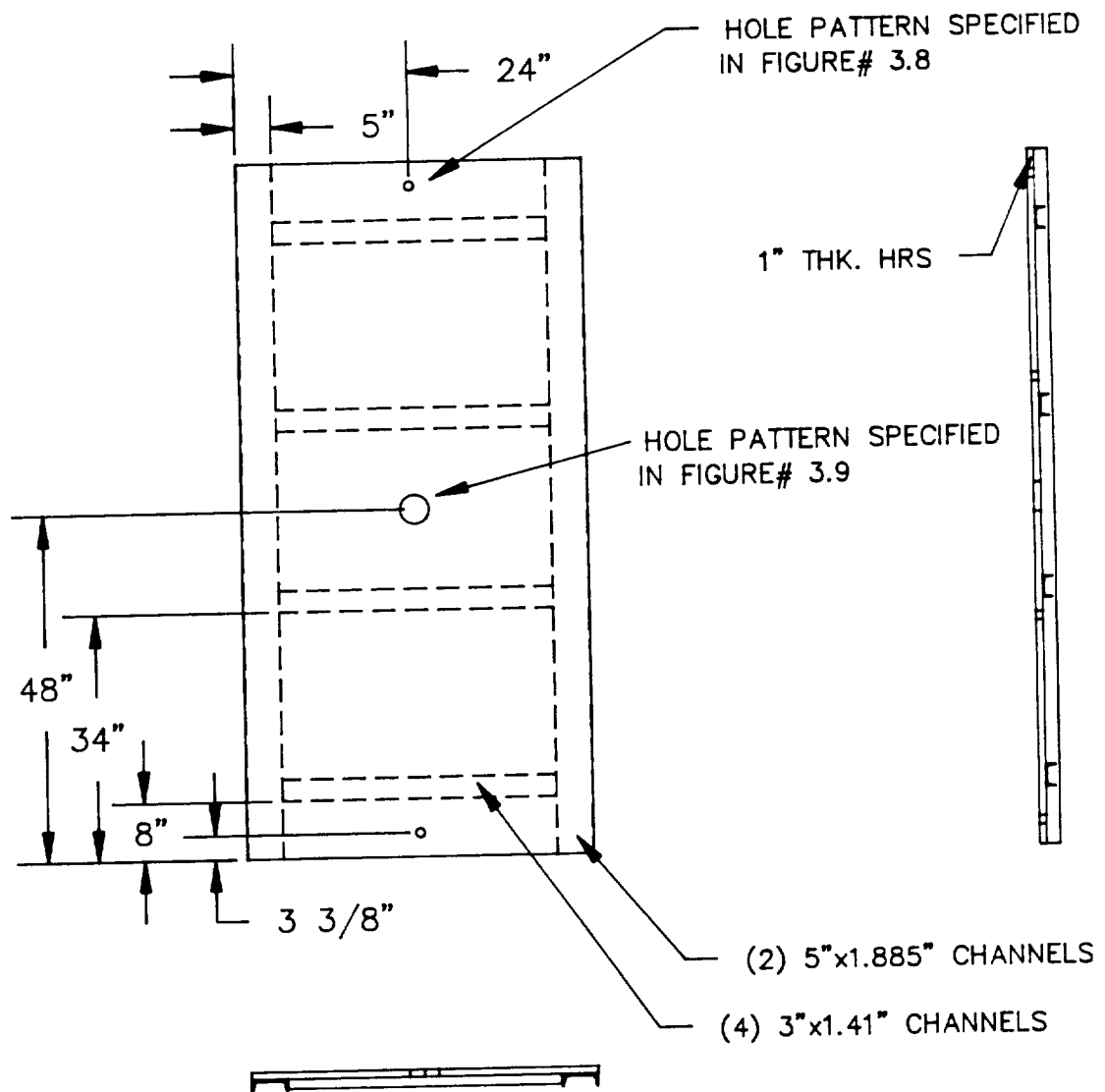
FIGURE #

3.6

DATE: 27 NOV 88

DESIGNER FNV

WELDED BELLOWS



#### NOTES

ALL CHANNELS TO BE WELDED WITH A 1/2" FILLET AT JOINTS, AND AT AN INTERVAL OF 2" EVERY 3" ALONG THE 1" THK. PLATE

GEORGIA INSTITUTE OF TECHNOLOGY  
ME 4182 DESIGN GROUP 7 - LUNAR DRILL

SCALE: 1/2"=1'

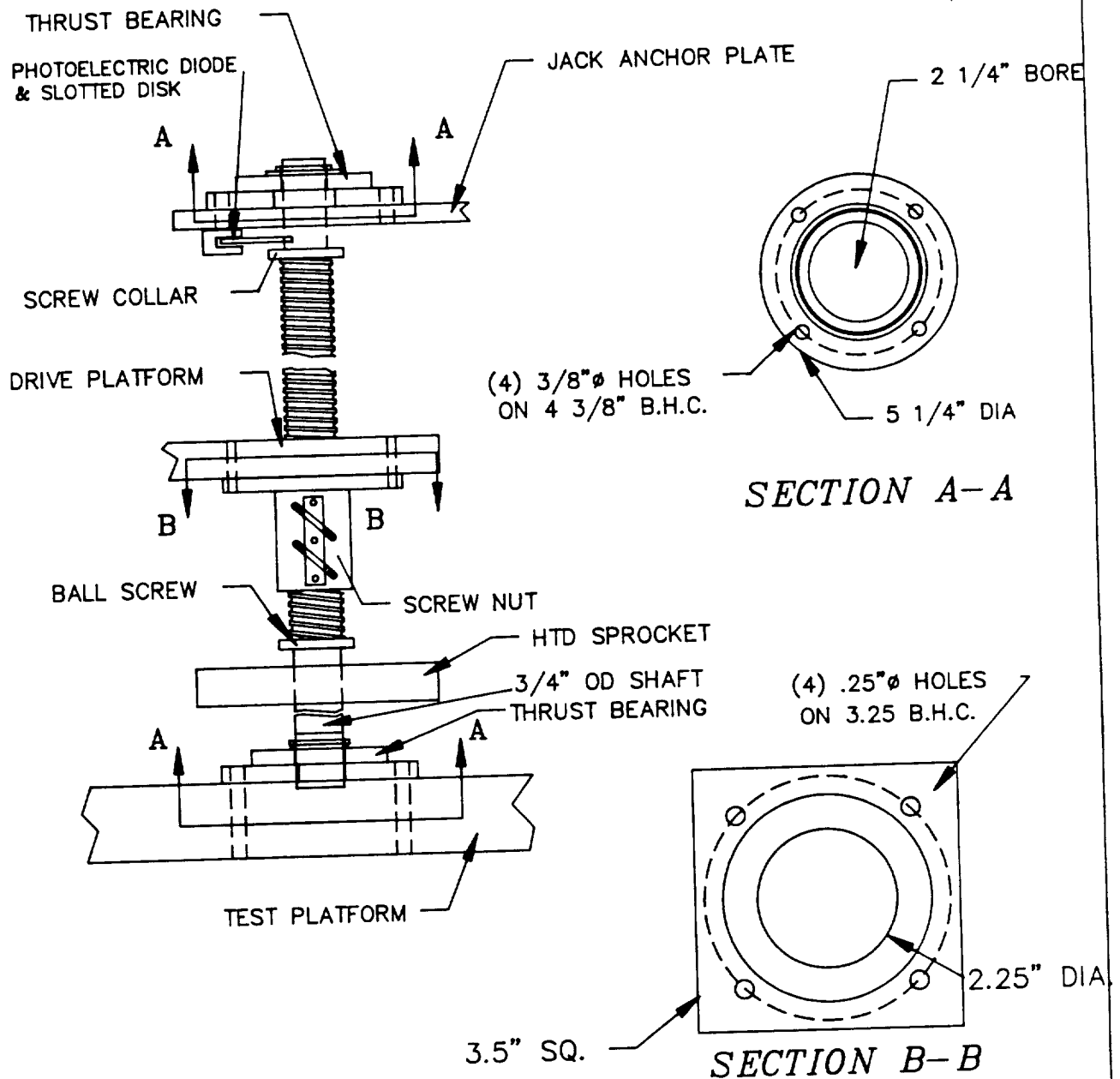
FIGURE #

3.7

DATE: 25 NOV 88

BY: FNV

DRIVE PLATFORM  
ASSEMBLY



GEORGIA INSTITUTE OF TECHNOLOGY  
ME 4182 DESIGN GROUP 7 - LUNAR DRILL

SCALE: 1/4"=1"

FIGURE #

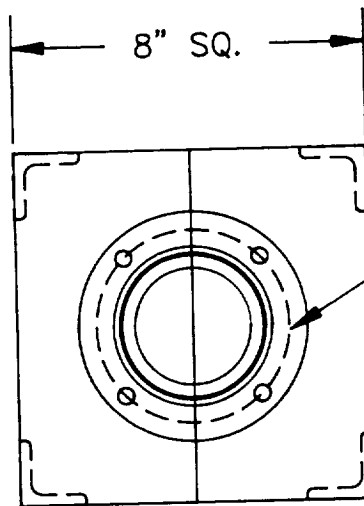
3.8

DATE: 25 NOV 88

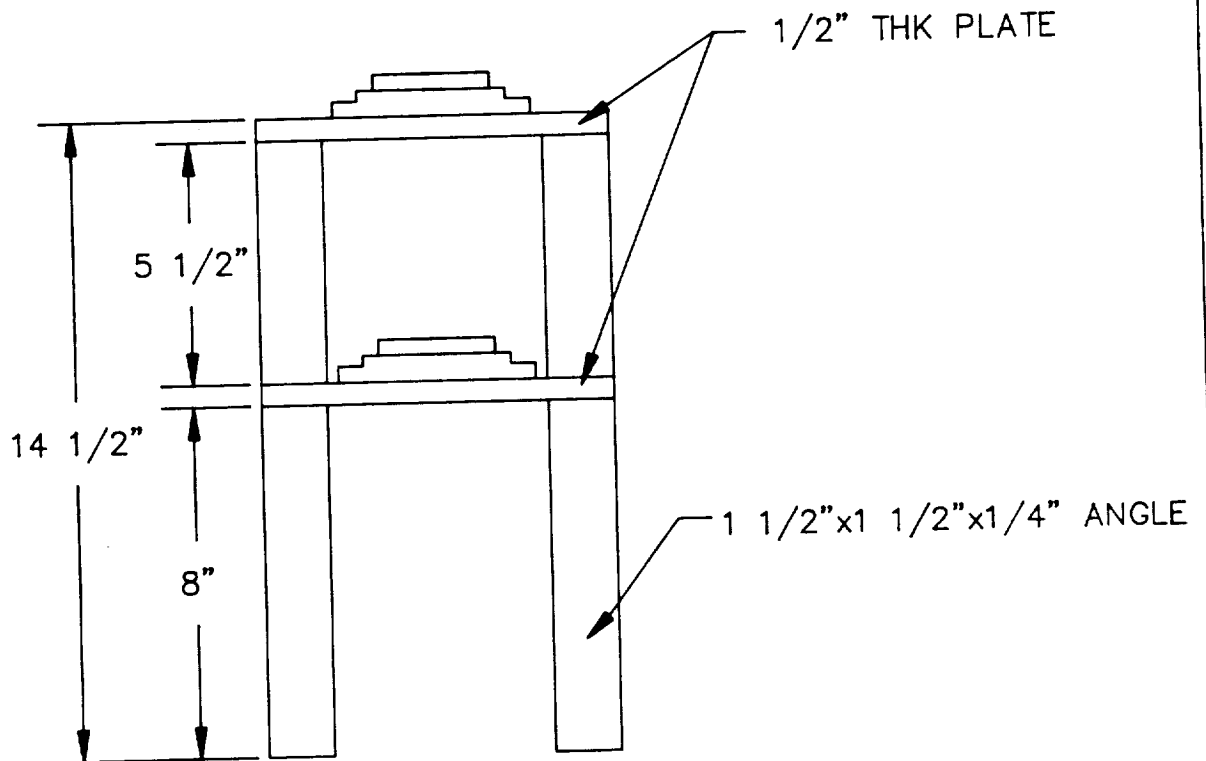
FNV

BALL SCREW JACK  
ASSEMBLY



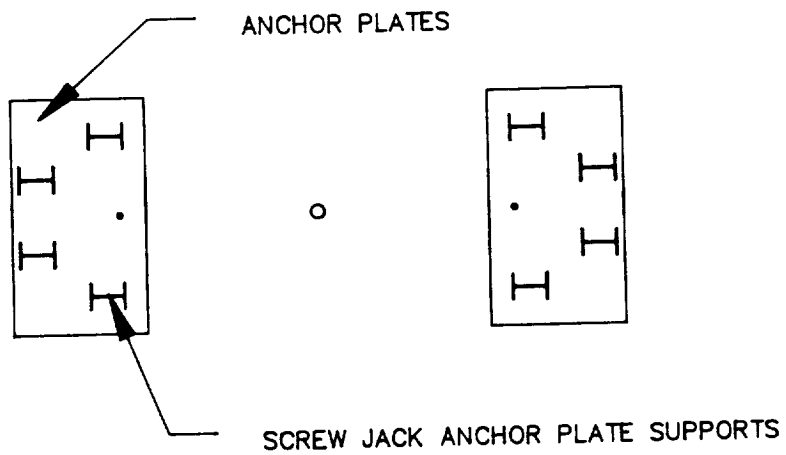


BEARING WITH BOLT  
PATTERN SPECIFIED IN  
SECT. A IN FIG# 3.8

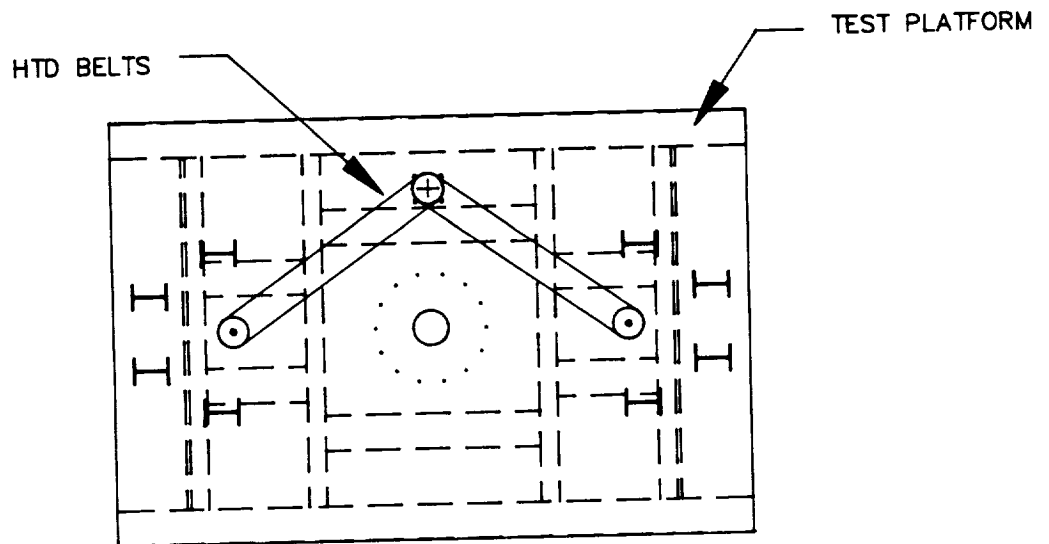


GEORGIA INSTITUTE OF TECHNOLOGY		
ME 4182 DESIGN GROUP 7 - LUNAR DRILL		
SCALE: 1/4"=1"	FIGURE #	3.9
DATE: 27 NOV 88	BY: FNV	

DRILL BEARINGS  
& MOUNTING



SECTION B-B



SECTION A-A

GEORGIA INSTITUTE OF TECHNOLOGY  
ME 4182 DESIGN GROUP 7 - LUNAR DRILL

SCALE: 3/8"=1'

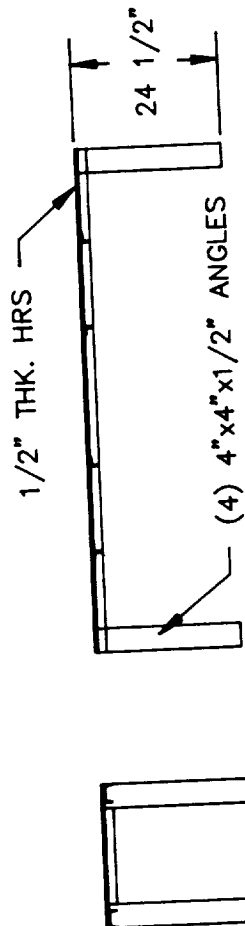
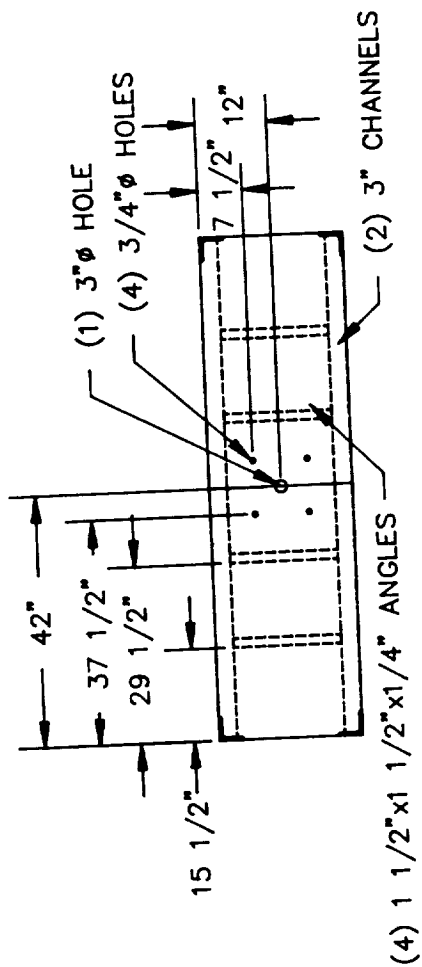
FIGURE #

3.10

DATE: 25 NOV 88

FNV

SECTION VIEWS



NOTES:  
ALL CHANNELS AND ANGLES TO BE WELDED WITH  
A 1/2" FILLET AT JOINTS, AND AT AN INTERVAL  
OF 2' EVERY 3' ALONG THE 1/2" THK. PLATE.

GEORGIA INSTITUTE OF TECHNOLOGY  
ME 4182 DESIGN GROUP 7 - LUNAR DRILL

SCALE: 1/2" = 1'

FIGURE #

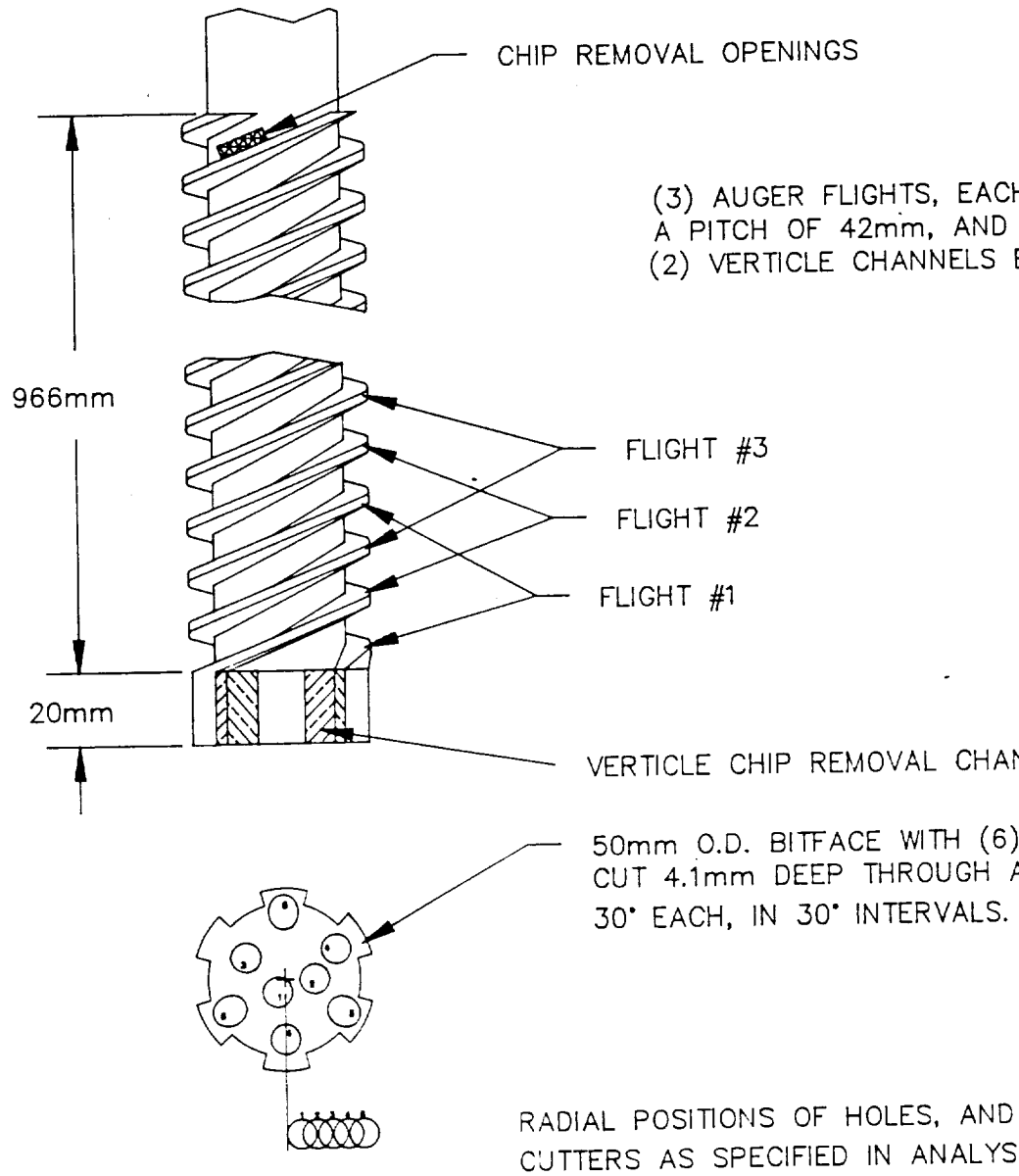
3.11

DATE: 27 NOV 88

FNV

MOTOR PLATFORM

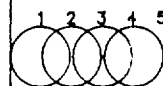
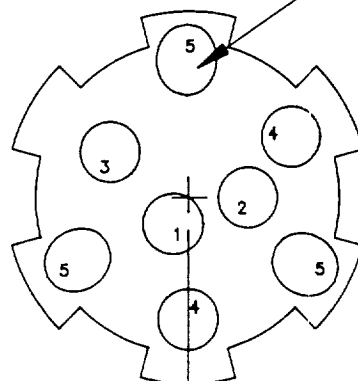
# FOLDOUT FRAME



# FOLDOUT FRAME 2

4 HAVING  
FED BY  
BELOW.

HOLES BORED 30° INWARD  
ALL OTHERS BORED STRAIGHT IN



BIT PROFILE

DETAIL A  
FULL SCALE

NEL

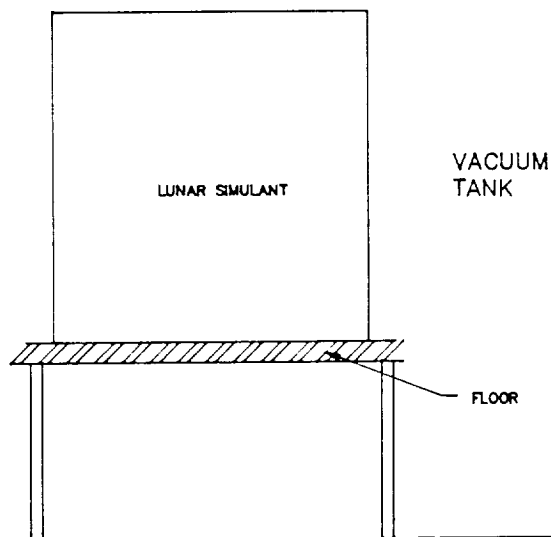
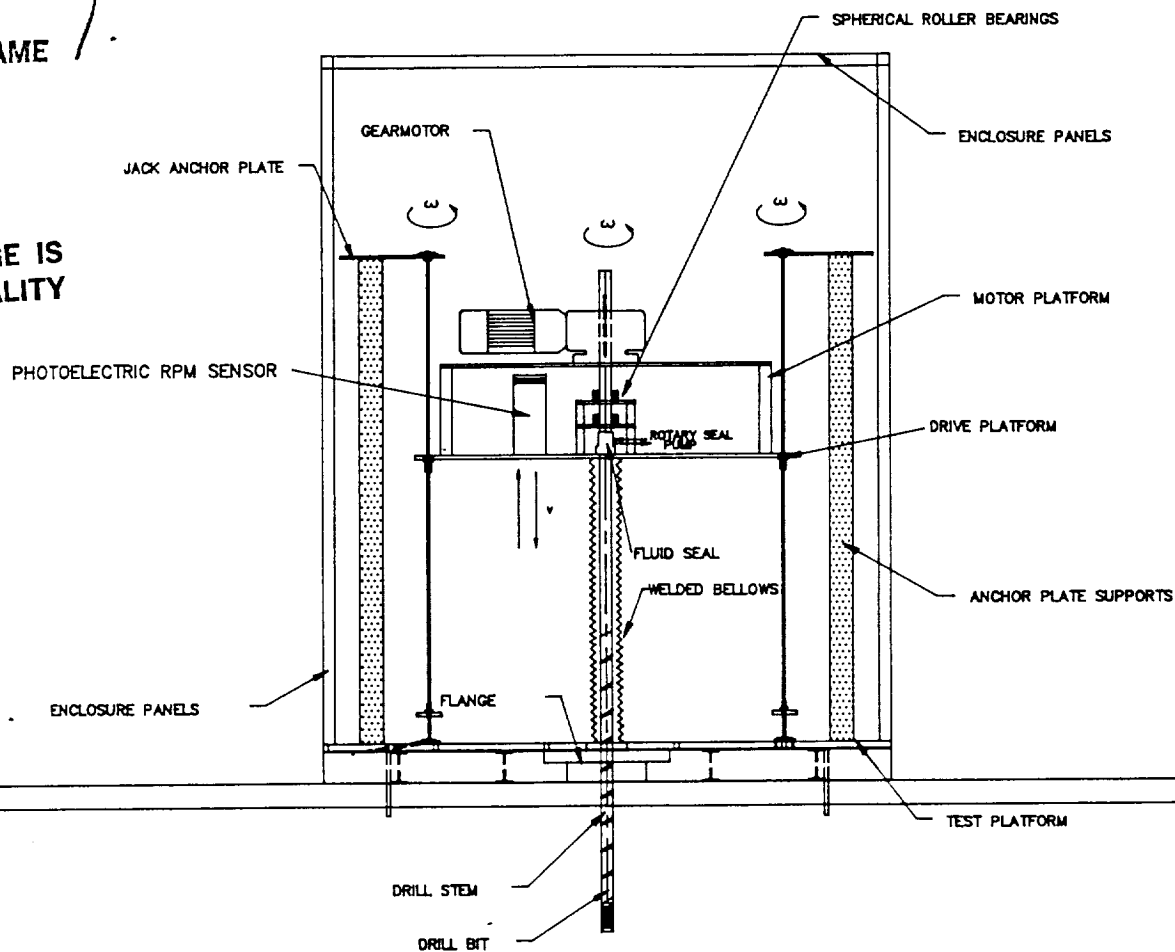
CHANNELS  
ANGLE OF

5.

GEORGIA INSTITUTE OF TECHNOLOGY ME 4182 DESIGN GROUP 7 - LUNAR DRILL			STUD LOCATIONS & AUGER DETAIL
SCALE: 1/2"=1"	FIGURE # 1.3		
DATE: 26 NOV 88	DRW: FNV		

FOLDOUT FRAME

ORIGINAL PAGE IS  
OF POOR QUALITY

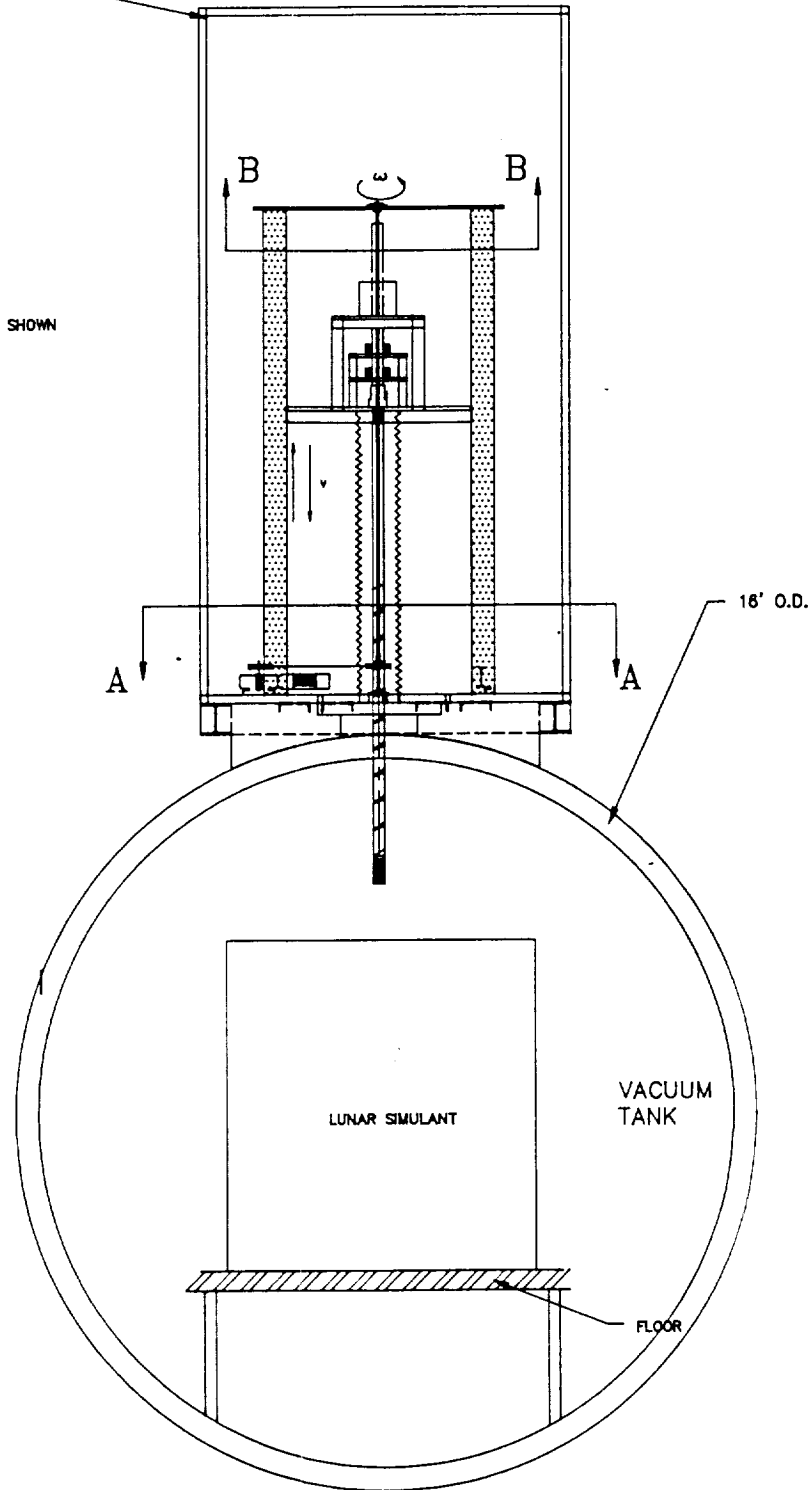


FOLDOUT FRAME 2

12'x8'x17' ENCLOSURE

ORIGINAL PAGE IS  
OF POOR QUALITY.

SECTIONS A-A AND B-B ARE SHOWN  
IN FIGURE 3.10



GEORGIA INSTITUTE OF TECHNOLOGY  
ME 4182 DESIGN GROUP 7 - LUNAR DRILL

SCALE: 1/4"=1'

DATE: 28 NOV 88

FIGURE # 3.1

DRAWN FNV

DRILL TESTING FACILITY

**The George W. Woodruff  
School of Mechanical Engineering**

**Georgia Institute of Technology**  
Atlanta, Georgia 30332